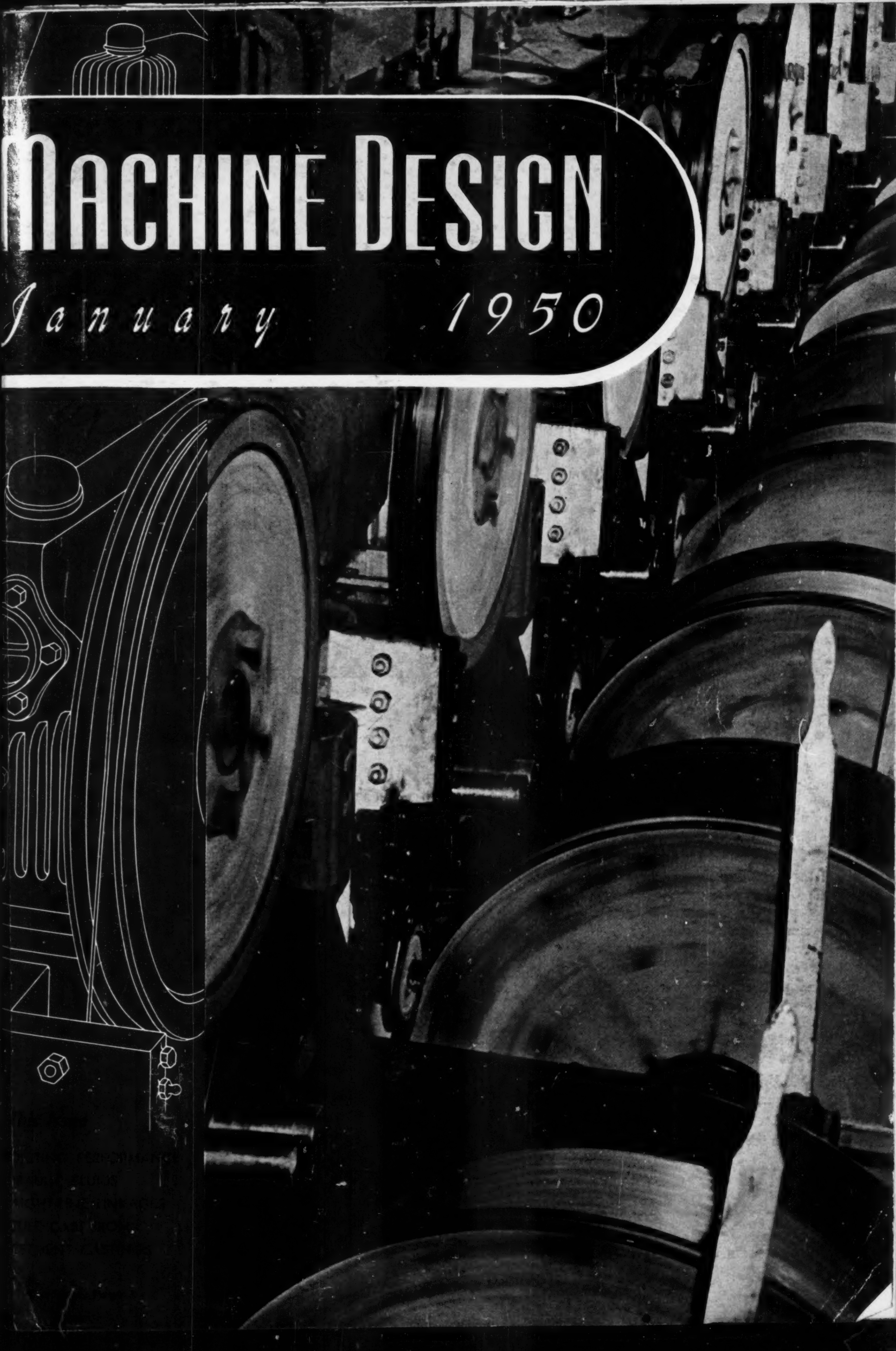
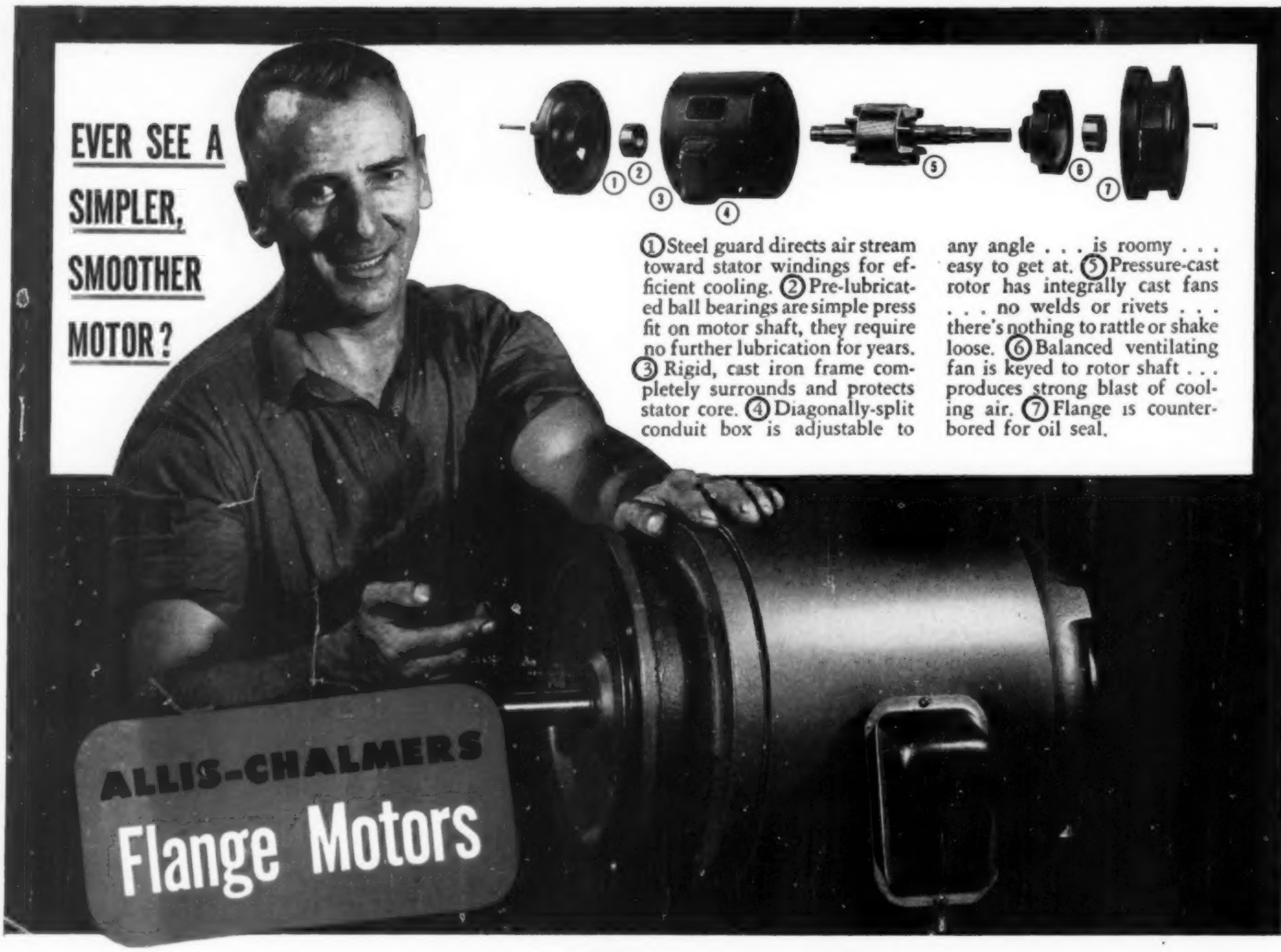
MACHINE DESIGN

January 1950





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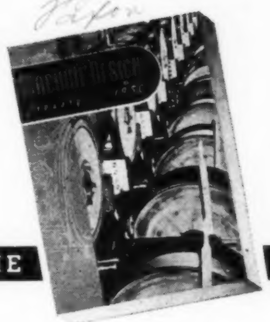

MACHINE DESIGN

Vol. 22—No. 1 **January, 1950**

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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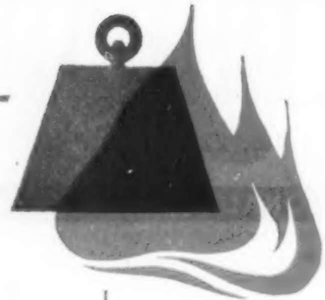

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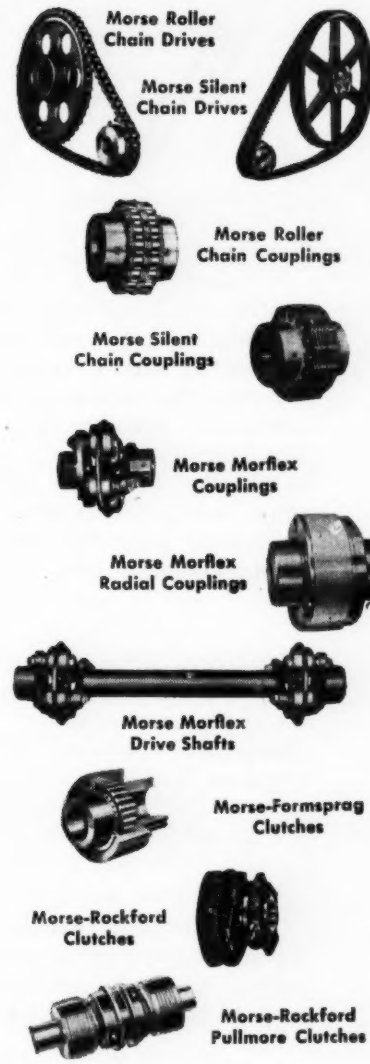

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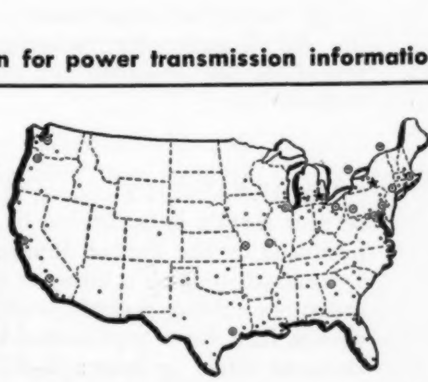

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TOPICS

LOW-CARBON type of 18-8 stainless steel with less than 0.03 per cent carbon has been developed by the Carnegie-Illinois Steel Corp., obviating the customary practice of adding columbium or titanium to the steel as stabilizers. Test samples of this new stainless steel have been boiled in 65 per cent nitric acid for 240 hours without corroding between the grains. Mechanical properties are similar to the higher carbon grades of 18-8.

RADIOACTIVE TRACERS are being employed at the National Bureau of Standards to study the mechanism of electroplating. With radioactive chromium 51 tagging either the trivalent or the hexavalent chromium ions in a chromic acid plating bath, it can be shown that the deposit is from the hexavalent state.

FIBER-GLASS PIPE, which may partially replace metal pipe, is tough, light and noncorrosive. Made of sand and synthetic resin it is easy to manufacture though costly at present.

TIMING MARKS every 0.001-second on high-speed camera film may be made with a device invented at the Naval Ordnance Laboratory. A battery-operated fork oscillator drives a six-channel power cathode follower which excites neon lamps in the cameras.

SUPERSONIC PROPELLERS for aircraft are being developed experimentally, according to the Air Force's Air Materiel Command. The propellers are designed for possible speeds up to Mach 1.5.

MACHINABILITY testing of ferrous and nonferrous metals is facilitated by the use of a unit produced by the Vanton Equipment Corp. Using a standard tool on a lathe or planer, the device measures cutting resistance in pounds per 0.001 square inch for a standardized chip area. Abrasiveness effect is determined by measuring the abraded flat on a 10-mm test ball brought into contact with the work material for a given time and pressure. Product of the two measurements is known as the Schlesinger machinability index and has promise of developing into a valuable universal standard.

BRIGHT SURFACE for metals is obtained without mechanical or electrical operations by a new process developed by Battelle Institute. The metal part is merely dipped into a chemical solution. When withdrawn a few minutes later it is polished to a high mirror-like luster.

SURFACE FINISHES may be analyzed readily with the aid of a liquid casting plastic developed by Marco Chemicals Inc. The plastic is poured over the surface to be checked and solidifies in about 15 minutes, giving a faithful replica of the surface.

METALLIZED PAPER for use in capacitors, originally developed in Germany, obviates the use of alternate layers of foil and paper.

ULTRASONIC ENERGY, high - frequency sound waves, can be concentrated in small areas where it is difficult to get other forms of energy. According to Norman F. Barnes, General Electric engineer, one kilowatt or nearly one horsepower can be concentrated into an area a centimeter square.

KENTANIUM, a material developed by Kennametal for high-temperature applications, will withstand extreme thermal shock while under tensile stress. In a test setup, the material withstood 600 cycles during 100 hours of alternately heating to 1800 F and cooling to 300 F in 15 and 30 seconds, respectively, while subjected to a tensile stress of 12,500 psi.

SYNTHETIC MICA, developed by the office of Naval Research, can be made in bulk form and readily machined to form many electrical insulator parts.

PUNCHES capable of piercing metal the thickness of which is as much as twice the diameter of the punch have been developed by the Pivot Punch and Die Corp. A "whipsleeve," pressure cast and shrunk to the punch where it tapers into the shank, supports the fracture point of the punch, strengthening the punch by transferring some of the stresses to the larger portion of the shank. The sleeve also serves as a guide in the stripper, minimizing vibration.

JANUARY, 1950

MACHINE DESIGN
A PENTON PUBLICATION

Facts or Guesswork?

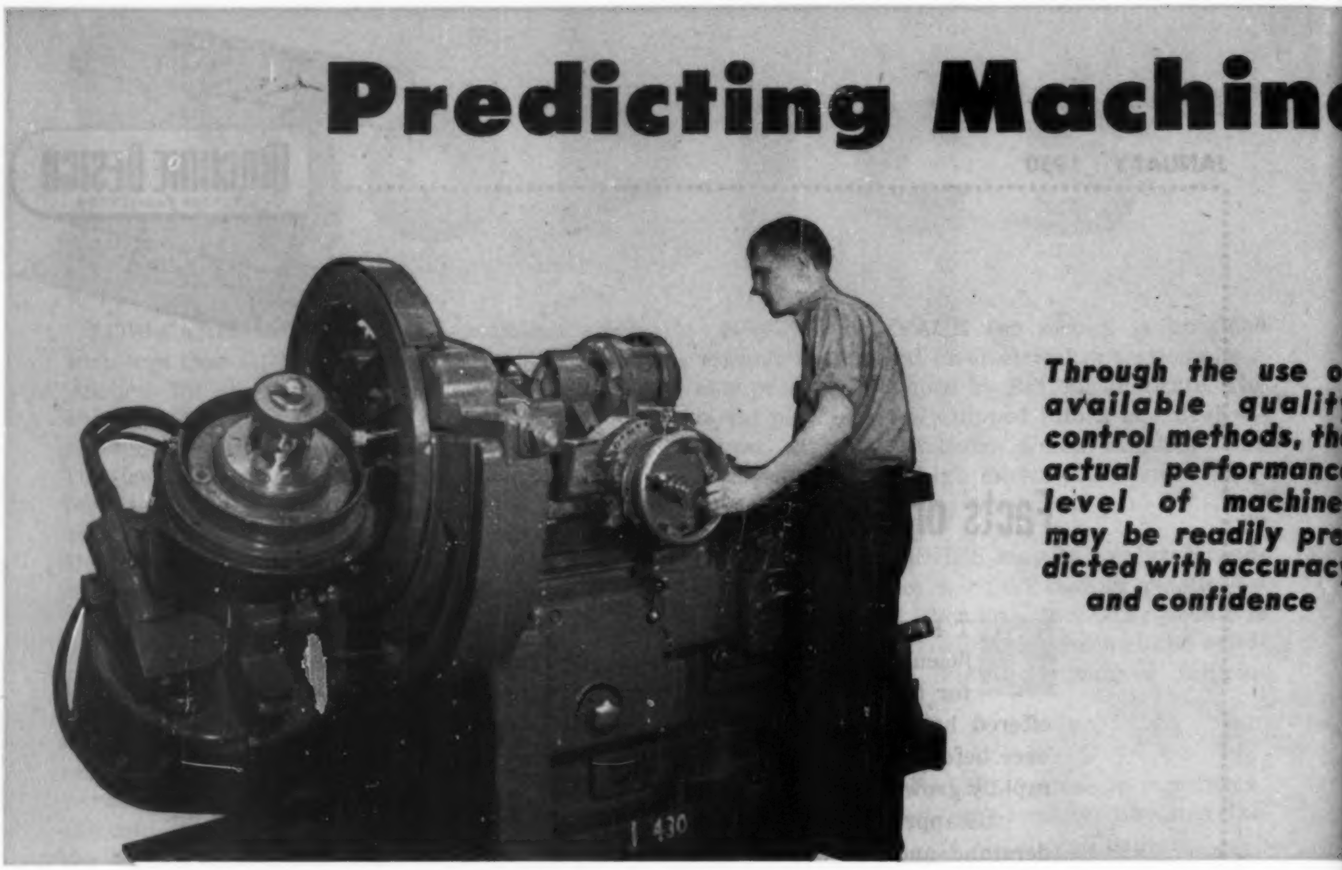
IT HAS been stated that among engineers only 10 per cent are proficient in higher mathematics. Perhaps this fact accounts largely for the failure of many to profit from the multitudinous advantages offered by the various phases of statistical analysis. Today, more than ever before, the engineer cannot afford to overlook the possibilities of this rapidly growing science.

Misapprehension that only the "long-haired" research analyst can understand and use these methods must be eliminated. For, once the engineer has opened his mind to this study the utter simplicity of the whole concept is apparent. Engineeringwise, it provides a tool of many facets. Far from being merely an inspection and production method of quality control, it gives a design and testing yardstick which will eventually become indispensable.

Even in simplest possible form, the use of these methods of analysis can be of everyday practical value. To help remove the veil of doubt MACHINE DESIGN, more than four years ago, presented the rudiments of this science to show how necessary refinements in the design of a machine could be pinpointed and determined. Prooftesting a new design, prior to shipment of the machine, to assure the designer that his judgment as to requirements was correct can save many production headaches, avert costly redesign and insure customer approval. In the current issue the Editors take pleasure in presenting another article which develops this interesting subject further.

Statistical methods also provide data on production operations which entirely eliminate individual opinions and make possible a sound practical basis for establishing design limits and fits. Guesswork in the establishment of fits and limits which are practical and economical, as well as in the achievement of satisfactory machine functions, cannot long be tolerated in the present state of development of the machinery manufacturing field. Statistical methods of analysis will prove to be an important key to better and more economical design.

ASSOCIATE EDITOR



Predicting Machine

Through the use of available quality control methods, the actual performance level of machines may be readily predicted with accuracy and confidence

Fig. 1—Above—A statistically analyzed machine can be confidently run by the customer—the "bugs" are out

TOO often production of a new machine is started or the shipment of the first few units is made after running the prototype to discover and eliminate obvious "bugs". Generally, the altered machine merely passes the test of an "expert" observer's opinion. The degree of failure of such a procedure is directly measured by the number and type of complaints and further changes that have to be made *after* the first units are in service—and even sometimes measured by the fact that your competitor seems to be getting the new orders.

Simple statistical techniques are available to bring to almost the vanishing point such customer problems. At this point one might understandably smile at the idea that one could explain in one article a universal technique that can find the bugs on any type of new machine that would escape the background knowledge, experience and intuition of an expert of many year's standing who has specialized on a particular type of equipment. Just to avoid discrediting the whole idea at this early point, consider this analogy. In the days before all drawings carried tolerances, and before micrometers were considered standard items of measuring equipment, expert fitters were heavily relied upon. Later it became evident that a high-priced fitter couldn't produce as accurate and as interchangeable a job as a relatively new man with a mike. The human or opinion element has been aided by factual measurement information. Again, it is logical to expect that a man with a microscope can offer some very revealing information about materials to people who have worked for years with those materials, but without a microscope.

Analogies, however, should not be used to reach an impractical conclusion. The experts are still useful and necessary. Give them an extra tool, one that replaces intuitive information with numerical charts, one that magnifies minor effects, and they then can more readily and effectively translate their knowledge and experience into the corrective action required.

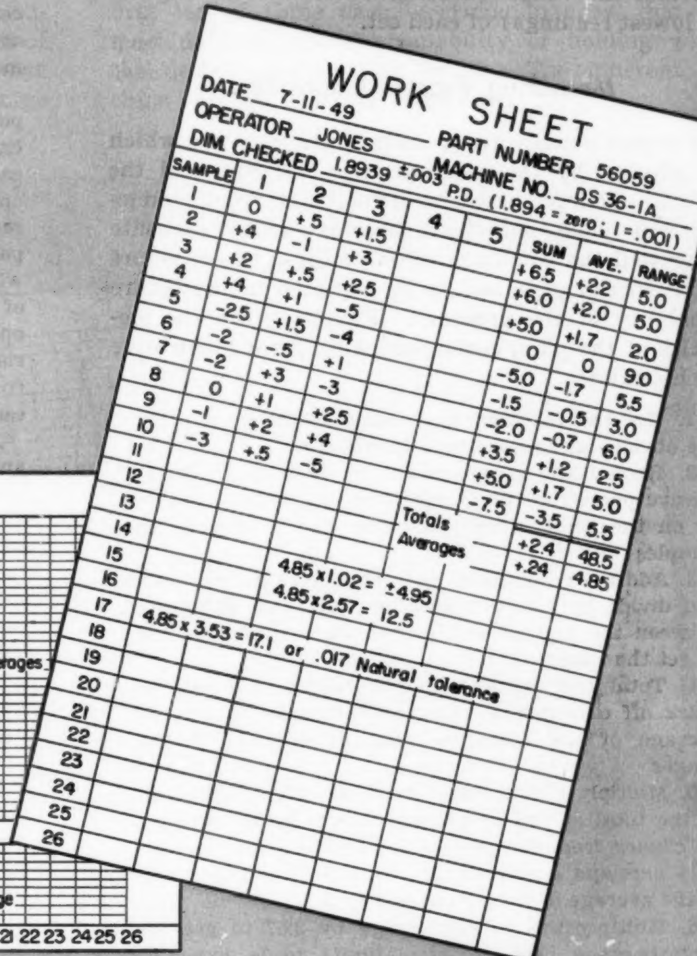
A machine will reveal its own condition provided some way is set up to measure the results of what the operation is supposed to do, such as turn, broach, grind, bore, or cut-off to a certain dimension; fasten something together with a certain strength; fill a container to a certain weight; plate to a certain thickness; or give a certain surface finish; and so forth. In order to translate measurements obtained into useful analytical information, they should be handled as detailed in the subsequent discussion.

Measurements usually represent the results of chance combinations of innumerable independent variables inherent in a process. While a study of the successive individual readings can yield some information about the capability of the process, trends,

Capability

By Dorian Shainin

Chief Inspector
Hamilton Standard Propellers
Div. United Aircraft Corp.
East Hartford, Conn.



WORK SHEET

DATE 7-11-49

OPERATOR JONES PART NUMBER 56059

DIM. CHECKED 1.8939 ±.003 P.D. MACHINE NO. DS 36-1A

(1.894 = zero; 1 = .001)

SAMPLE	1	2	3	4	5	SUM	AVE.	RANGE
1	0	+5	+1.5			+6.5	+2.2	5.0
2	+4	-1	+3			+6.0	+2.0	5.0
3	+2	+.5	+2.5			+5.0	+1.7	2.0
4	+4	+1	-5			0	0	9.0
5	-2.5	+1.5	-4			-5.0	-1.7	5.5
6	-2	-.5	+1			-1.5	-0.5	3.0
7	-2	+3	-3			-2.0	-0.7	6.0
8	0	+1	+2.5			+3.5	+1.2	2.5
9	-1	+2	+4			+5.0	+1.7	5.0
10	-3	+.5	-5			-7.5	-3.5	5.5
11				Totals			+2.4	48.5
12				Averages			+.24	4.85
13								
14								
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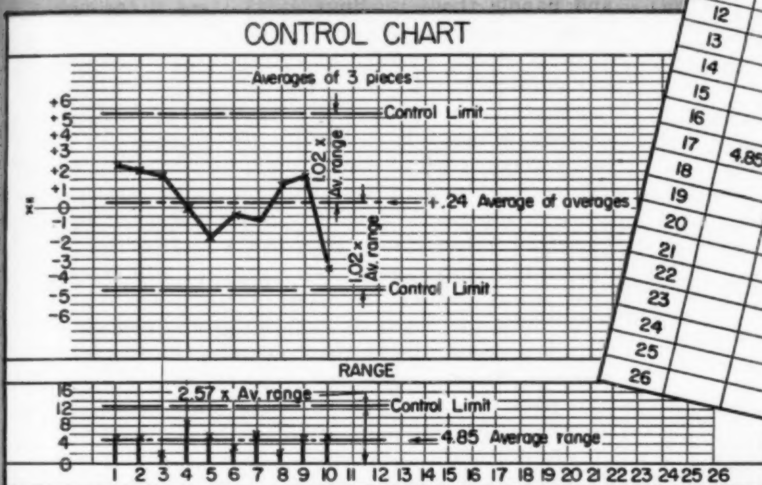
4.85 x 1.02 = ±4.95

4.85 x 2.57 = 12.5

4.85 x 3.53 = 17.1 or .017 Natural tolerance



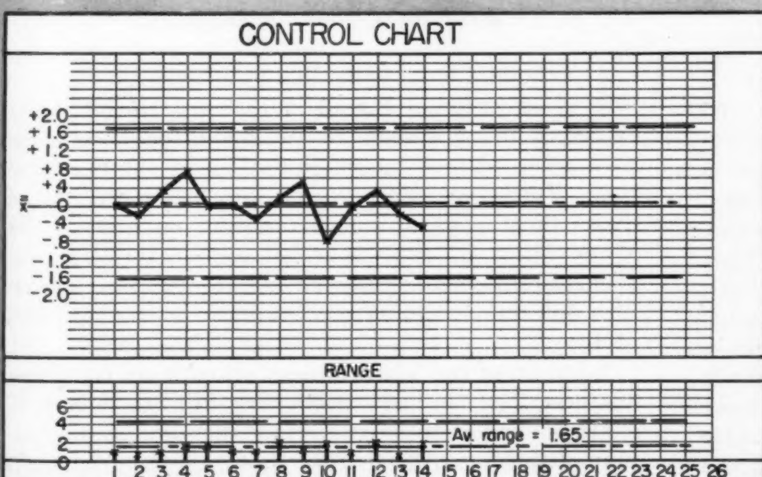


Fig. 2—Above—Thirty individual successive readings are simply tabulated in sets of three each. The values in the "Ave." and "Range" columns are plotted

Fig. 3—Above—Left—Ten samples of Fig. 2 plotted (construction details shown). All points being within the control limits mean there are no assignable causes of erratic variation, although the inherent variation exceeds that specified

Fig. 4—Left—Same machine of Fig. 3 is now producing to a 0.0058-inch natural tolerance (0.00165 x 3.53). This was reduced from 0.017-inch by redesign

effects of trial changes, etc., this type of analysis can often mask desirable information, be sometimes actually misleading, and always will require a great deal more data to reach reasonable conclusions than the special statistical methods which will be recommended. These newer methods are more effective simply because the laws of chance, defined by the mathematics of probability, can assign definite odds to certain *combinations* of numbers regardless of how unpredictable the individual numbers might be. Thus, the technique employed consists in comparing the actual combinations with the expected. If they agree, the laws of chance are operating (the process can only be improved by a basic change such as design). If they do not agree, there are one or more assignable causes of extra variation operating erratically which usually can be rather readily eliminated.

The *combinations* which will be discussed are two

simple ones—*averages* of each of several sets of readings and the *range* (difference between the highest and lowest readings) of each set.

Measuring Process Capability

Processes usually give results which vary and which are affected by the operator, the material, and the machine. Thirty successive items taken from production will yield sufficient information to estimate rather well the combined effect of these three factors on the spread of values to be expected in the future (capability of the process). Following are the mechanics of the plan, accompanied by three illustrative case histories:

1. Get the machine in running order by making the obvious and necessary adjustments
2. Record the measurement of each of thirty successive parts run off. Keep these in groups of three, as on the Work Sheet shown in *Fig.* 2, giving ten samples
3. Add the readings of each sample to get the sum and divide by 3 to get average. Take the difference between the highest and lowest value in the sample to get the range
4. Total the ten averages. Total the ten ranges. Mark off one decimal point on each total to get the average of the averages, and the average of the ranges
5. Multiply the average range by 1.02 to get half of the total spread of *average values* to be expected *by chance* from the process. This figure will be used as a plus and a minus value *control limit* either side of the average of the averages, *Fig.* 3
6. Multiply the average range by 2.57 to get the largest *range value* (control limit) to be expected *by chance* from the process
7. Plot the ten average and the ten range points as in *Fig.* 3. Draw the three control limit lines.

If all the points are within the control limits, the characteristics of the process which are causing the variation in results are operating steadily. There is nothing erratic about the results, there are no *assignable causes* of variation which can be readily found. In fact, when this situation exists, you can multiply the average range by 3.53 to get the *natural tolerance* of the process—the total variation to be expected in the *individual* measurements. If this tolerance is too great to meet specification requirements, you will know that the only way to reduce it is to change some feature of the basic process (usually the machine design). An inherent variable needs to be reduced or eliminated.

On the other hand, if some of the points are beyond the control limits, then an assignable cause of erratic (more than chance) variation can usually be found and eliminated rather readily. It can be found in the operator, the material, or the machine (which includes the gaging device which gave the measurements).

Finding Machine Troubles

In those rare instances when an assignable cause is indicated and is not easily found, the general procedure outlined in the folowing will unearth it:

8. List all suspected causes of erratic variation, e.g., nonuniform chucking action, erratically sticking cam, hardness variation of material, nonrepeatability of gage reading, varying amount of stock to be removed, etc.
9. Devise a method to measure each of these suspected causes, e.g., an appropriately mounted indicator will show the runout of the work as chucked each time, or the position of a cam at its critical operating point at each cycle, a hardness tester will reveal the variation of that physical property in the parts processed, using one piece of work repeatedly with the gage will show the gage variation, a series of micrometer or indicator measurements before the operation will show the different amount of material to come off each piece (using unprocessed pieces from the same source as the thirty already processed), etc.
10. Take thirty readings of each suspected cause, and handle these readings as in steps 2 through 7 in the foregoing procedure
11. If all the average and range points are within the control limits, cross out that suspected cause from your list. A characteristic that is running or varying in accordance with pure chance will normally have a pure chance effect on the results of the process, and thus it will not be an assignable cause of erratic variation. If all the suspected causes are thus crossed from your list, you have not suspected the right one. Guess again, or call in other experienced personnel to guess
12. If all the points are not within the control limits then, more likely than not, that cause is assignable. Take appropriate steps to bring its variation back into a chance pattern, and then recheck your final desired product to see if the evidence of all assignable causes has now disappeared.

Machine Clamp Redesigned

A difficulty was being experienced with the cutting of an external thread in that the thread miller employed did not seem to be able to hold variation within the plus or minus 0.003-inch of the specification. Thirty successive pieces were measured with a dial indicator gage, and the results entered on the Work Sheet of *Fig.* 2. The sums, averages, and ranges were computed for each three successive readings and the results were plotted on the control chart of *Fig.* 3.

Addition of the control limits, which were computed as described in the foregoing, to this control chart immediately reveals that the process is running within control but that its inherent variation is too great for the specification requirements. As is shown on the Work Sheet of *Fig.* 2, multiplying the average range by the factor 3.53 shows that the natural tolerance of this process is 0.017-inch, against the specification value of 0.006-inch.

The machine designer in this particular instance, therefore, was on the lookout for a major cause of inherent variation in his machine that required redesigning. He found that the clamping mechanism utilized to retain the head in its preset position was far too fragile to resist the loads imposed upon the head by the cutter. Clamping the head merely produced a bending action in the mechanism, the frictional holding power developed by the clamp being

very small, thus permitting an inherently wide variation from the set position.

This condition was remedied by redesigning the entire clamp arrangement. The resultant production of the next thirty pieces gave the control limits shown in *Fig.* 4. The multiplication of the average range by the factor 3.53 now gives a natural tolerance of 0.0058-inch. The four additional samples, beyond the first ten of three each, continue to show that this machine now has the capability of holding within the desired specification limits. The inherent machine variable has been eliminated.

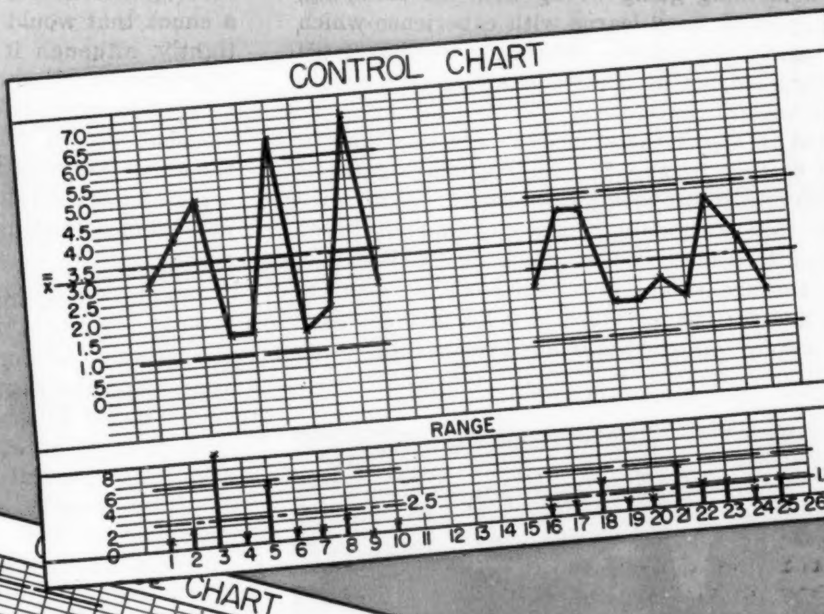


Fig. 5 — Right — "Before" and "after" record of chucking problem. Any one of the three out-of-control points would indicate the presence of an erratic cause of unnecessary variation. Cleaning out a sticking chuck gave the right-hand results which now truly show the capability of the machine

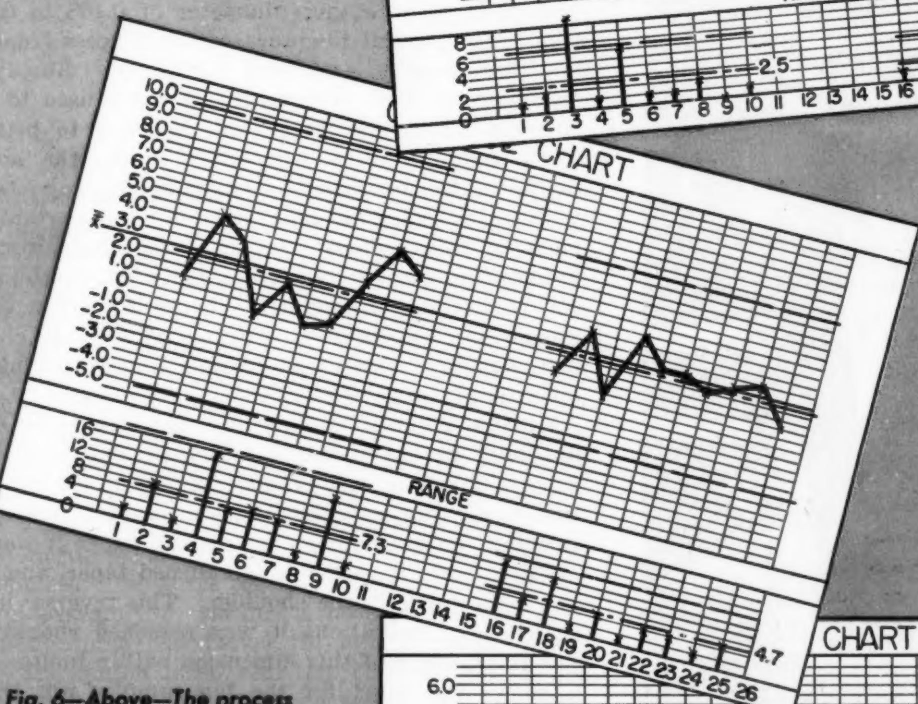
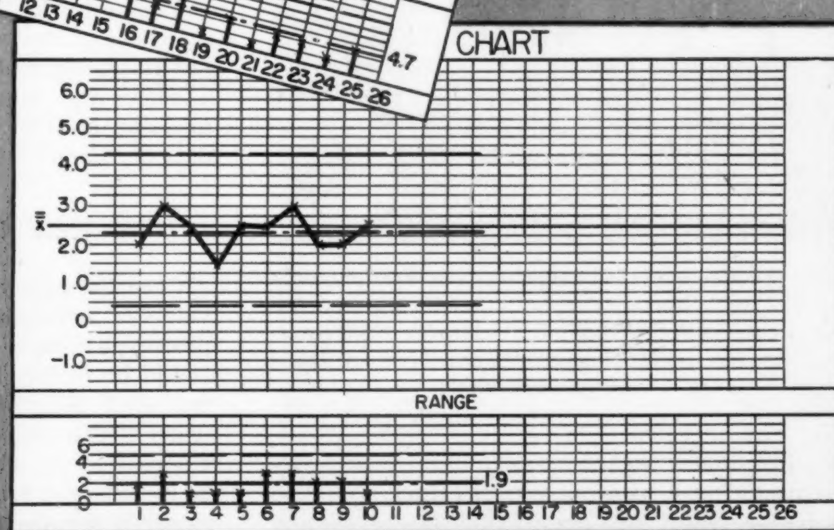


Fig. 6—Above—The process variation often consists of more than just the contribution of the machine. When the human element is involved, the in-control variation can sometimes be narrowed by giving the operator a better gage to guide his settings

Fig. 7—Below—The natural tolerance of the lathe of Fig. 6 was too great to hold the part tolerance specification of 0.008-inch. Here is a machine with a 0.0067-inch natural tolerance (0.0019 x 3.53)



An operator now need only continue such a chart with these control limits and watch for the first point that may in the future fall outside one of them. Such a situation will more than likely be found with a sample of three pieces that are still all within the drawing limits but which have an average or range value that shows that something not characteristic of past results is now happening. This something could well be a dull cutter, a misadjustment in the machine, something going wrong with the clamping, etc. The operator soon learns with experience which of these easily correctable difficulties is correlated with the type of out of control situation he encounters on the control chart.

Note that in *Fig.* 4 the scale has been blown up as compared to that of *Fig.* 3. An important point is thus illustrated. The scale used must be of such magnitude that the results can be clearly seen. When the natural tolerance was as much as 0.017-inch, a scale for average values of a half a thousandth for each line and for range values of 0.002-inch for each line was ample. On the other hand, as natural tolerance of the process was decreased it became necessary to use a finer measuring device and each line on the average chart now stands for 0.0002-inch, and the range scale utilized became 0.001-inch for each line.

Fig. 8—By the clever use of indicators, one can not only measure the variations characteristic of any suspected cause of erratic variation, but also determine the "capability" or general performance level and "natural tolerance" of the whole machine

Erratic Chucking Action

The left-hand side of *Fig.* 5 represents another type of difficulty found in a machine design. Here, the first ten samples showed one range point and two average points out of control. The machine, therefore, was not operating with a variation characteristic only of its design in that these out of control points were evidence that something was very erratic. An investigation revealed that the assignable cause was a chuck that would not always hold the work equally tightly, although it seemed so to the operator. The right-hand portion of *Fig.* 5 shows the results of ten successive samples after the chuck was taken apart, cleaned up, and reassembled. It is a matter of interest to note that the machine setting had been dropped from a value of $3\frac{1}{2}$ units to $2\frac{1}{2}$ units in the process of correcting this machine.

Machine Gaging Important

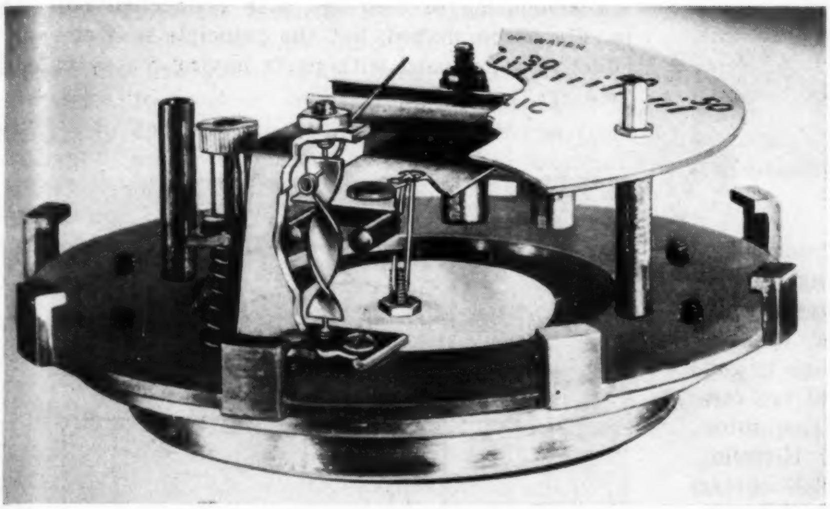
A three part story of machine design capability is represented in *Figs.* 6 and 7. *Fig.* 6 shows an attempt to use a lathe to hold a shoulder position on a ground taper to a gage diameter of 0.168 to 0.176-inch. The natural tolerance of the process from the first ten samples was over 0.025-inch. Simply by changing the type of gage the operator used to control his work, this figure was reduced to between 0.016 and 0.017-inch. In other words, the human factor of the operator's reading of the gage in adjusting his lathe for each piece is an important element of this process. In each case there were no assignable causes present, i.e., there was nothing erratic about the process, the inherent variation in the second run having been reduced by a basic change in the gaging which was a guide for controlling the operation from piece to piece.

At this point it was decided that the lathe could not hold to the required tolerance and a different type of machine was needed. Fortunately, the solution here was not extremely involved. Instead of forming the shoulder to the ground taper it was decided to form it to the rough turned taper, and then grind the taper to the shoulder. This reversal in the sequence of operations it was reasoned should permit the holding of this dimension within limits. *Fig.* 7 shows the chart for the ten samples run on the grinding machine. The natural tolerance from these data is computed as 0.0067-inch.

Proof Test Machine Prototypes

Once the machine designer recognizes the practical value of determining the natural tolerance of each prototype design from running thirty pieces, *Fig.* 8, it can be readily seen that he can confidently predict the level of machine performance. It follows logically that a copy of such a chart should be furnished the user of the machine. Such a report of the quality to be expected from the output of the machine is certainly a service that the manufacturer can well afford to give his customers in return for some excellent good will and confidence.

SCANNING the Field For Ideas



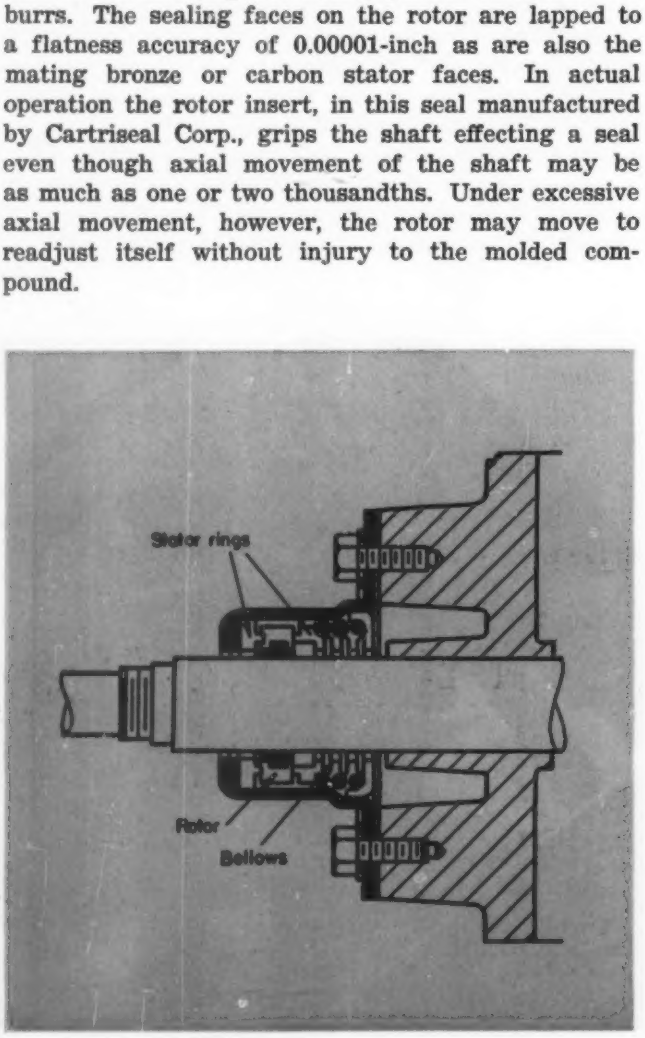
Magnetic linkage between diaphragm and indicating pointer in the pressure gage illustrated in cutaway view, above, makes the instrument sensitive to the slightest pressure change. Designed by F. W. Dwyer Manufacturing Co., the gage utilizes a permanent magnet to translate the motion of a flexible diaphragm into rotary motion of the pointer. The diaphragm is supported by a cantilever spring on which is mounted the magnet. A flat strip twisted in the shape of a helix forms the shaft of the pointer and is located between the poles of the magnet in such a way that vertical motion of the magnet results in rotary motion of the shaft. This action is frictionless, the force of the magnet causing the helical assembly to assume a position in which the magnetic force can travel through a minimum air gap. The action is so sensitive that motion of the magnet as small as 0.0003-inch are reflected in minute movements of the pointer.

Precise corrections for errors in time intervals, measured by commercial timers, are indicated and recorded on the frequency-error indicator illustrated at right. The momentary deviations in frequency in city power lines cause corresponding errors in interval timers because they are driven by synchronous motors. The correction system which depends on the frequency-error indicator has an accuracy of 0.01-second, extending the usefulness of interval timers to the realm of precise measurements.

The cumulative error, arising from the variations in the frequency of the power supply driving the timer, is obtained over the period of observation by comparing the speeds of the two black synchronous motors shown in the photograph. One is driven by the power line and the other by a standard crystal-controlled frequency. Any difference in the speeds of the two motors results in motion of the differential gear mounted on the shaft between them. Thus, the position of the gear shaft at any instant depends on the total accumulated frequency error in the power line. The shaft is connected to indicating dials which are calibrated to give the errors in seconds and hundredths of a second. A transmitter selsyn on the other end of

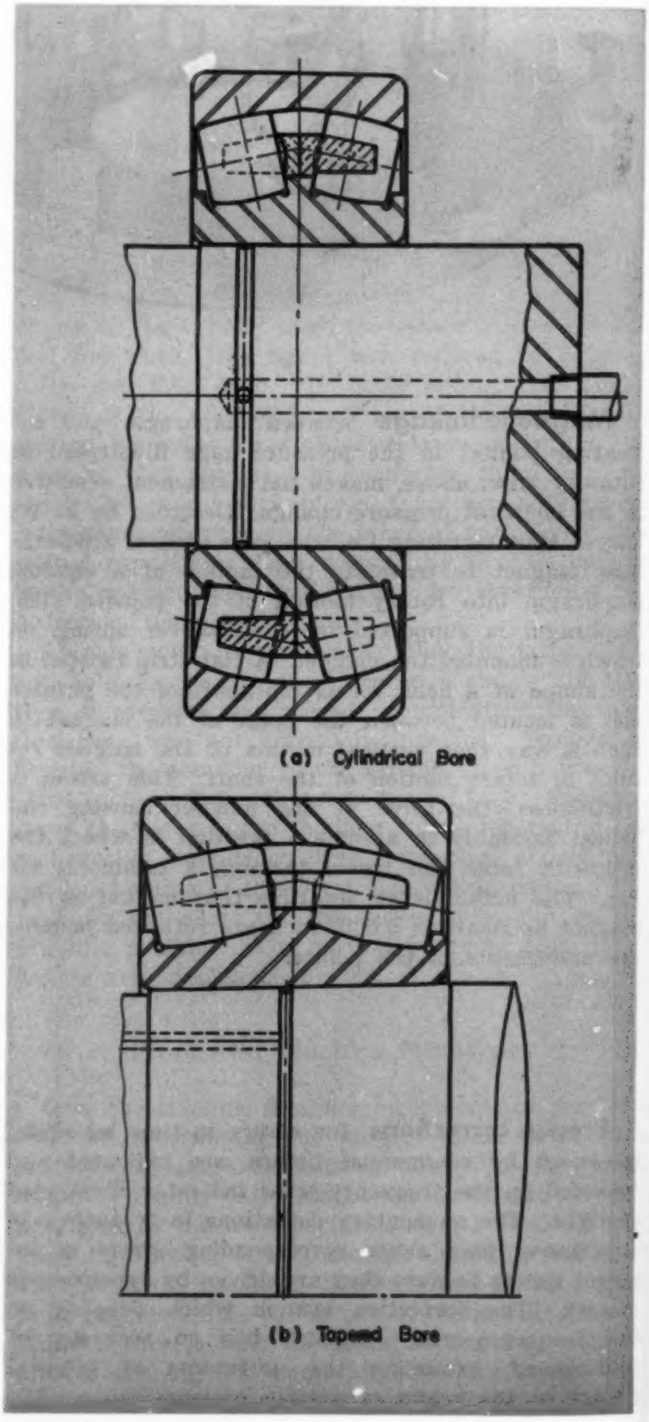
the shaft transmits the angular motion of the differential gear shaft to a receiver selsyn which operates a recording unit. The power amplifier in the foreground amplifies the crystal-controlled frequency for driving the standard synchronous motor in this instrument designed by R. E. Gould and H. A. Bowman at the National Bureau of Standards.

Unit assembly of a rotary seal into a cartridge, below, eliminates the usual precision fitting on machines and provides for easy replacement in the field. As designed for a refrigerator compressor, the seal unit is slipped onto the shaft and the flange is gasketed to the housing. All parts of the seal are contained in the cartridge including stator rings, rotor, bellows and spring. The rotor is either Nitralloy or Malcomized stainless steel with a special Neoprene insert. This insert fits snugly on the shaft, a rough turned finish being sufficient, just so there are no burrs. The sealing faces on the rotor are lapped to a flatness accuracy of 0.00001-inch as are also the mating bronze or carbon stator faces. In actual operation the rotor insert, in this seal manufactured by Cartriseal Corp., grips the shaft effecting a seal even though axial movement of the shaft may be as much as one or two thousandths. Under excessive axial movement, however, the rotor may move to readjust itself without injury to the molded compound.



Oil-pressure method for mounting or dismounting large bearings and other interference fits virtually eliminates friction between contact surfaces, making the operation a matter of minutes and protecting the parts against unintentional abuse. Developed by SKF Industries Inc., the system involves introducing oil from a hand operated pump between the shaft and the bore of the bearing so that the ring is expanded to "float" on the oil film. Friction is reduced to one or two per cent by oil injections which seldom exceed a pressure of 5000 psi.

Dismounting of bearings with cylindrical bores is readily accomplished, but the principle is even more effective when used with parts having mated tapers.



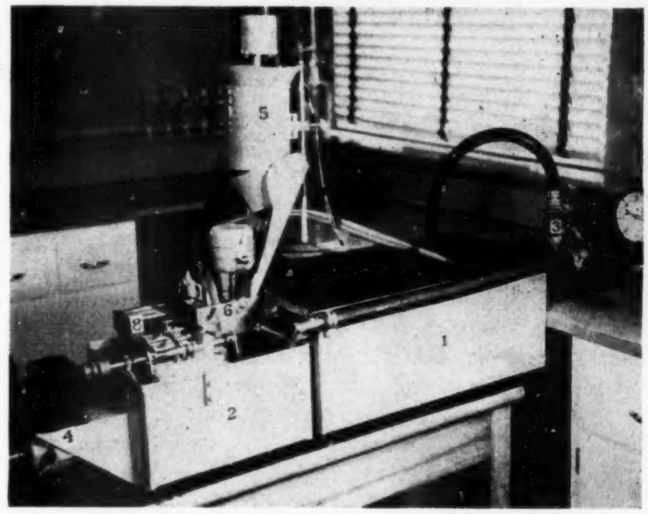
(a) Cylindrical Bore

(b) Tapered Bore

The arrangement for a cylindrical bore is shown at *a*, at left. The oil port—a radially drilled hole connected to a peripheral groove in the shaft seat—is close to one side of the bearing. This off-center location is in the same direction as the motion when stripping a bearing from its seat. Despite the port location, however the bearing usually will not come completely off its seat because the oil film cannot be maintained by small areas. A small force is needed to remove the bearing the remaining short distance of the fit.

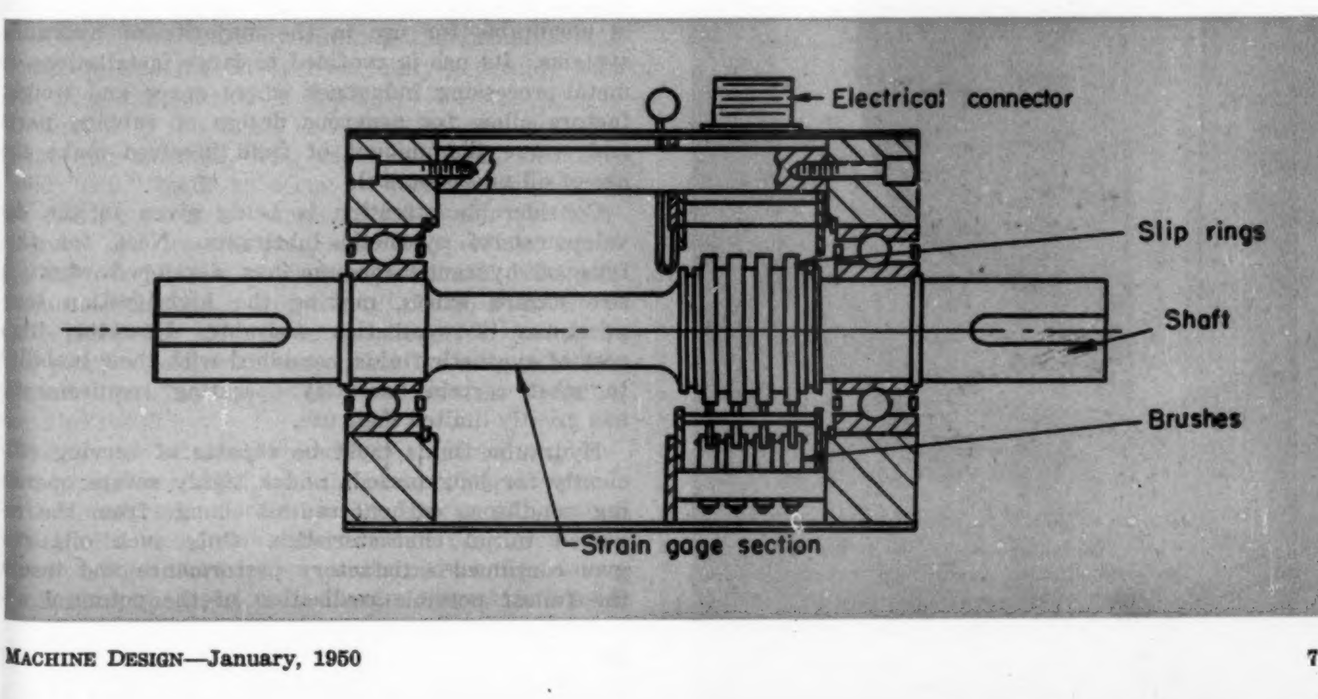
Arrangement for dismounting a taper-bore bearing is shown at *b* for a standard 1 to 12 taper. When oil pressure is applied to the bore, a safety stop is necessary to prevent overtravel. In mounting a taper-bore bearing, the component axial force from hydraulic pressure must be taken into account. Actual measured force to accomplish assembly is usually 20 to 30 per cent greater than the theoretical force calculated to effect the displacement.

Strain gages, mounted in the form of a Wheatstone bridge on the reduced section of the shaft in the instrument, below, are employed to measure torques applied to the shaft. Designed by Baldwin Locomotive Works to use SR-4 bonded resistance-wire gages, the instrument has the grids of the gages placed at 45 degrees to the longitudinal axis and connected so that the effects of bending and thrust stresses are automatically cancelled while the effects of torsion stress are cumulative and cause a proportional bridge unbalance. Corners of the bridge are connected to four insulated slip rings through which voltage is supplied to the bridge and its output is measured. This type of instrument has been found useful for torque measurement in automotive and airplane engines as well as in a variety of other applications such as handwheels, pumps, compressors, and fans.



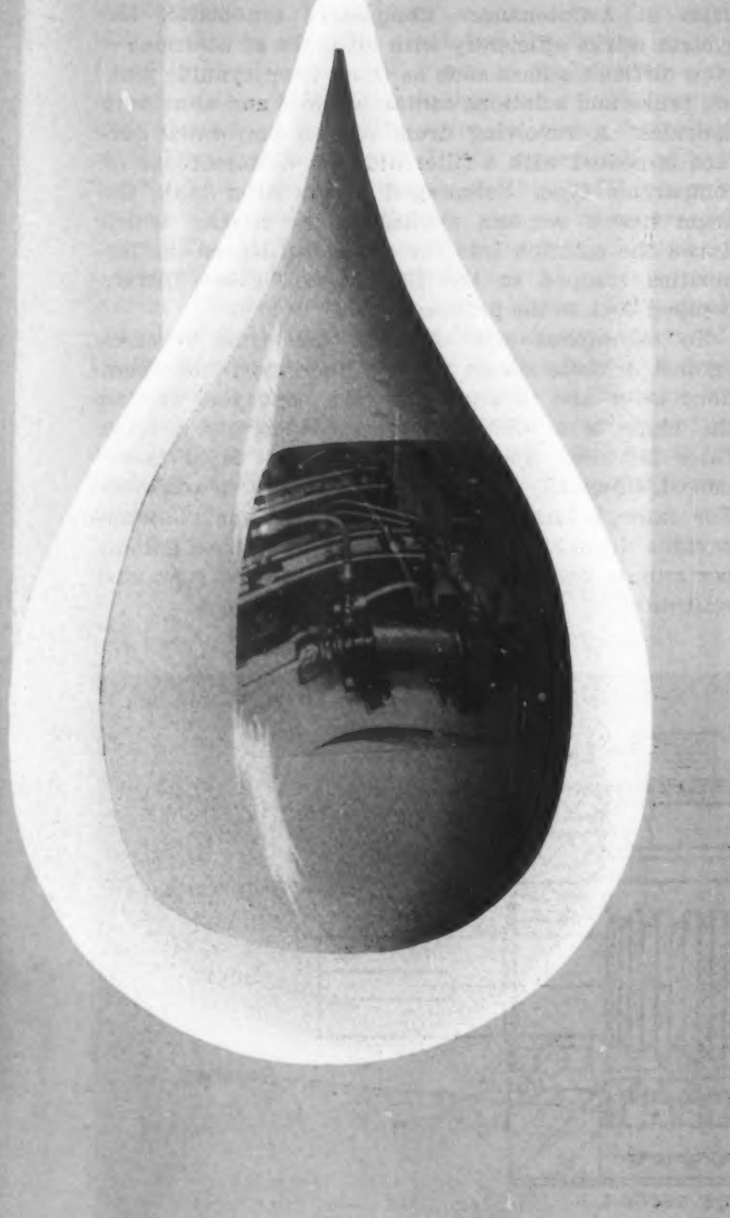
Self-renewing filter system, above, developed experimentally by the Commonwealth Engineering Co., has high oprating efficiency and requires a minimum of maintenance. Completely automatic, the system works efficiently with all types of solutions—even difficult slimes such as those from cyanide plating tanks and solutions containing iron and aluminum hydrides. A revolving drum with a perforated surface is coated with a filter aid of a diatomaceous or comparable type. Submerged in the filter tank, the drum has a vacuum applied to its interior which draws the solution into the drum and leaves the impurities trapped in the filter cake. Clear filtrate is piped back to the process.

To accomplish self cleaning, the drum revolves against a blade set at the top to remove the spent filter layer and to deposit it on a conveyor. Behind the blade is a slurry chamber, depositing a fresh filter layer on the drum as the spent layer is removed. Unusually high flow rates have been achieved. For example, using aqueous solutions, the pilot apparatus illustrated has reached a flow of five gallons per minute per square foot of filter during sustained operation.



Give LIFE to Hydraulics

. . . by proper selection of oil and careful design of components



HYDRAULIC oil must be considered as an essential component of hydraulically controlled machines and should be selected with the same careful thought given to other components which comprise a modern machine. This discussion will be devoted primarily to petroleum oil as a hydraulic-fluid medium, covering the essential characteristics of hydraulic oils, the deteriorating influences to which they are subjected in service, the potential effects of these influences on the service life of the oil, and the characteristics required of an oil for satisfactory performance. Also, mechanical factors affecting the satisfactory performance of hydraulic fluids will be considered.

Much has been said in recent years about the outstanding weaknesses of hydraulic design from the users' viewpoint. Builders and their designers welcome such criticism as constructive and essential to continued progress in the improvement of hydraulic mechanisms. In this spirit this article points out weaknesses observed by lubrication engineers, which particularly affect hydraulic oil performance and makes suggestions as to improvements in design which should help to eliminate or, at least, to minimize some of the prevalent difficulties.

Although petroleum oils are not ideal hydraulic fluids they are, generally speaking, superior to anything else available today. Water, because of its lack of wear-preventive and rust-preventive qualities is unsuitable for use in the majority of hydraulic systems. Its use is confined to large installations in metal-processing industries where space and weight factors allow for generous design of rubbing parts and where the amount of fluid involved make the use of oil uneconomical.

Considerable attention is being given to the development of synthetic lubricants. Need for this type of hydraulic medium has developed where a fire hazard exists, making the high-ignition temperatures of synthetics desirable. However, high cost of synthetic fluids, combined with their inability to meet certain essential operating requirements, has greatly limited their use.

Hydraulic fluids must be capable of serving efficiently for long periods under highly severe operating conditions without radical change from the required initial characteristics. Only such oils can give continued satisfactory performance and insure the fullest possible realization of the potential ad-

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Systems

vantages of hydraulic systems. To fully appreciate this fact, it is necessary to consider the essential nature of hydraulic oils and the importance of each individual characteristic. This involves two broad considerations; first, the influence of various oil characteristics on the performance of hydraulic systems; second, the contribution of certain qualities toward resisting deteriorating influences encountered in service.

Viscosity: Viscosity of the oil must be such as to insure free flow throughout the system, with prompt response of controls and cylinders or fluid motors. An oil too heavy in body will flow sluggishly with consequent sluggish machine action and lost production. The relatively high resistance to flow results in high fluid friction which is overcome only at the expense of wasted power. The useless work is converted into heat as indicated by increased temperatures, and this heat accelerates oil deterioration. Under extreme conditions, such as low temperature at start up, the oil may be too heavy to flow freely into the pump suction. This results in cavitation, which is indicated by noisy pump operation and may result in damage to the pump. In intial phases, cavitation may be spasmodic, probably due to the thinning and relatively free flow of oil adjacent to the hot pump alternating with the much slower flow of more remote oil. This results in intolerably erratic machine action which can be overcome, short of changing oil, only by running the machine long enough to thin the oil by frictionally generated heat until proper operation is possible.

If an oil is too light, excessive leakage, both internal and external, may result. Internal leakage, called "slip," tends to occur between the stationary and moving parts of pumps and between the high-pressure and low-pressure sides of pistons, vanes, and valves. This is most important with variable-displacement pumps where any decrease in volumetric efficiency must be compensated for by lengthened pump stroke with consequent increase in power demand. Inability to develop full system pressure may also result.

The volume of oil delivered by a constant-displacement pump normally exceeds system requirements. In this case, slip merely results in less oil passing through the relief valve and there is no power loss. However, the fluid friction resulting from the forcing of oil through restricted passages causes increased heating, which not only thins the oil so as to increase slippage but also accelerates its deterioration.

Viscosity Index: In addition to suitable viscosity at operating temperatures a hydraulic oil should have a good viscosity index in order to minimize cold start-up troubles and reduce the length of "warm-up" periods. The graphs shown in *Fig.* 1, represent five hypothetical oils each having a viscosity of 150 SSU at 100 F, but differing in viscosity index as shown. In studying the points of main importance on these graphs it will be noted that the effect of varying viscosity indexes is not as great as might be anticipated, considering the emphasis which has been placed on the subject during recent years.

Referring first to the variations at 130 F, which is taken as the generally observed maximum bulk oil temperature in hydraulic systems under normal operating conditions, it is obvious that the differences shown are of no practical importance for production machine systems. Considering next the

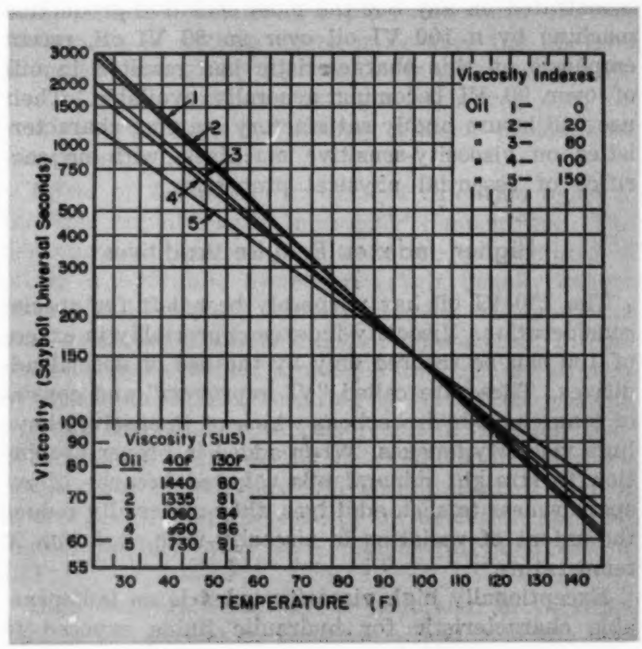


Fig. 1—Influence of viscosity index on viscosity at significant system operating temperatures

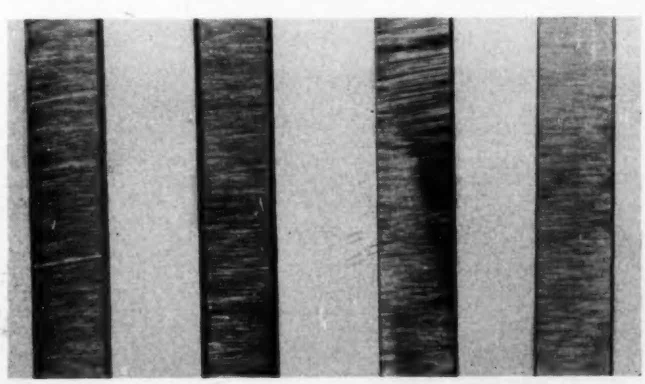


Fig. 2—Comparative wear of pump vanes. Shown, left to right, vane tested with straight mineral oil, two with additive oils, and a new vane. Runs were for 1000 hours under accelerated test conditions

lowest temperature at which the various oils reach the generally accepted maximum of 4000 SSU, it is noted that this does not occur even with the zero VI (viscosity index) oil until the temperature drops to 22 F. From this it can be assumed that any oil having a viscosity of 150 SSU at 100 F qualifies as to specified fluidity at the lowest temperature likely to prevail in a shop.

The effect of viscosity differences at 40 F, which is taken as the lowest probable start-up temperature in a production shop, must be considered in greater detail. The most practical basis for comparison is to consider the zero and 20 VI oils paired against those of 80 and 100 VI. There is much operating experience to show that the difference between low VI (zero to 20) and ordinary high VI (80 to 100) oils may mean an additional hour or more of warm-up time before acceptable operation can be secured. This applies particularly to high-speed internal grinders, diamond borers and similar rapid-cycling, high-precision, viscosity-sensitive machines. While it is questionable that any great benefit could be demonstrated on any but the most sensitive production machine by a 100 VI oil over an 80 VI oil, recent emphasis on this characteristic has resulted in oils of over 90 VI becoming generally available. Their use will insure highly satisfactory starting characteristics on viscosity-sensitive machines, with no sacrifice of essential physical properties.

Higher Indexes Require Additives

The 150 VI oil has purposely been left for special consideration. Viscosity indexes appreciably in excess of 100 can be secured only by the use of special additives. These are called "VI improvers" and consist of long chain hydrocarbons which of themselves have high viscosity indexes. When added in proper proportion to straight mineral oils selected because of responsiveness to such additives, they materially reduce the extent of variation in viscosity with variation in temperature.

Exceptionally high viscosity index is an indispensable characteristic for hydraulic fluids exposed to wide extremes of temperature variation, as in aircraft. However, there is an important consideration which prevents their widespread adoption for machine tool systems. The long chain hydrocarbon additives tend to break down when subjected to frequent passage through close clearances at high pressures over long periods of time. Under such conditions, they may be permanently reduced in viscosity. Such reduction may approach 30 per cent, carrying with it the danger of decreasing the viscosity of the entire oil batch below that required for safe operation. This consideration indicates that a viscosity index between 90 and 100 is about the practical limit for today's hydraulic oils for all types of production machines.

Pour Point: Considering the conditions under which hydraulic systems usually operate, the pour point of hydraulic fluids is of no practical concern. All oils now used as hydraulic fluids have pour points well below 32 F. Special oils are available for applications involving lower temperatures. An important precaution in this connection is not to rely on pour point alone for insuring adequate fluidity. Some heavy oils and those having poor viscosity indexes may have exceptionally low pour points, but their fluididity at low temperatures is actually comparable to "molasses in January." Correctness of viscosity for the pertinent temperature range and of viscosity index must both be considered along with pour point whenever the latter is a critical consideration in the system operation.

Film Strength and Lubricity: These two characteristics are so interdependent in minimizing friction and wear that they may be considered together. Before proceeding to define these terms, it is pertinent to consider the two general classes of lubricating films that may exist. Under favorable operating conditions and with an ample supply of a properly selected oil, a full-fluid film can be maintained. Such a film completely separates the relatively moving surfaces, eliminating metallic friction. The only friction to be considered is that due to resistance to shear within the fluid oil. Insofar as friction reduction is concerned, correct viscosity is the only characteristic required as long as full-fluid-film lubrication prevails.

Under heavy unit pressures, poor film forming conditions or scanty oil supply, full-fluid lubricating films cannot be maintained. Slight increases in pressure, shock loads, vibration or increased temperature tend to cause film rupture at the highspots on the rubbing

Fig. 3—Pump parts after 1000-hour run with high-quality straight mineral hydraulic oil A, Fig. 5. Note varnish-like deposits and score marks

surfaces, resulting in more or less extensive metal-to-metal contact. This general set of conditions is referred to as boundary lubrication and, when they prevail, the film strength and lubricity characteristics of the lubricant are highly important.

Film strength is the property of a lubricant that enables it to maintain protective films on the rubbing surfaces of machine parts under conditions of boundary lubrication, and is essential in order to prevent execessive wear, scoring and seizure of the parts. Lubricity, also referred to as oiliness or slipperiness, is the property of an oil which reduces the coefficient of friction under boundary film conditions, and so contributes to the smooth movement of rubbing parts. A proper combination of both of these properties is required in order to secure efficient results.

Adequate Film Strength Prevents Wear

If an oil does not have sufficient film strength to resist rupture under prevailing pressures, serious wear of pump elements and other rubbing parts is likely to occur. Wear of pump vanes or pistons and of valves results in increased clearances with the same effects as when too light an oil is used—viz, slippage, pressure loss, high temperatures and thinning of the oil, with further increase in wear—a vicious cycle which will do costly damage if not halted by mechanical adjustments or the use of an oil better suited to the prevailing conditions. Piston rods are also subject to wear. When badly worn or scored they make it virtually impossible to prevent leakage at piston glands.

It is fortunate that properly selected additives in double-inhibited oils (discussed later) materially enhance film strength, thus giving added protection against wear. Such additives also seem to have some beneficial effect as to lubricity. This, in addition to the inherent lubricity of suitable mineral-base oils, insures smooth operation of hydraulic systems in the majority of applications.

CHEMICAL STABILITY: This characteristic is undoubtedly the least understood property of petroleum oils as well as the most difficult to explain. It may be broadly defined as the ability of an oil to resist combination with oxygen to form deposits of gum or sludge. Oxidation and the consequent formation of such gums or sludges is one of the major sources of trouble in hydraulic systems. It is not within the scope of this article to present an involved discussion of the phenomenon of oxidation. However, it is pertinent to discuss briefly the major causes of oil oxidation in hydraulic systems, with emphasis on those which can be controlled, to some degree, through the co-operation of machine designers, builders, operators, lubrication, and maintenance engineers.

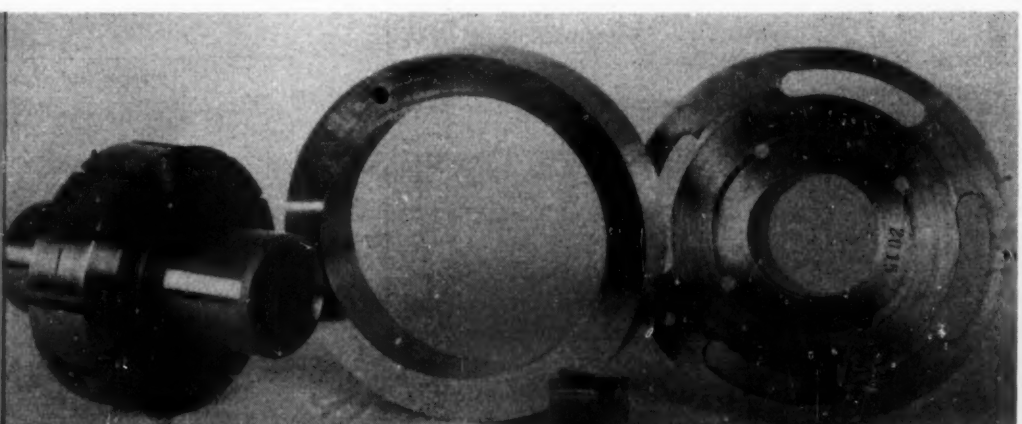
In order for oxidation to occur, the oil must have intimate contact with oxygen. This is facilitated by the tendency of mineral oils to dissolve air. The amount dissolved varies with temperature and pressure. For example, at 77 F, under atmospheric pressure, one gallon of oil will dissolve 23 cubic inches of air; at the same temperature but under a pressure of 200 psi, one gallon will dissolve 322 cubic inches of air.

The principal sources of air in hydraulic systems are: Entrainment by return oil dropping and splashing into the reservoir; entrance through the pump suction, either because of pin-point holes or poor connection in suction lines, or because the oil level drops low enough to uncover the inlet; careless addition of makeup oil, permitting a high drop with excessive splashing; and leakage around packings at low-pressure points.

Effect of Temperature: Rate of oxidation tends to increase rapidly with increasing temperature. Under normal operating conditions the temperature of oil in a hydraulic system reservoir usually ranges from 100 to 130 F. However, cycle characteristics may cause considerably higher temperatures. In constant-volume systems, prolonged dwell, short strokes, or other conditions resulting in much oil passing through the relief valve may result in reservoir temperatures as high as 180 or even 200 F. The importance of temperature will be appreciated when it is recognized that the rate of oil oxidation approximately doubles with each temperature increase of 18 F.

High temperatures may be generated in variable-volume pumps due to excessive slippage in the system, and also when the oil is subjected to frequently repeated reversals as in some duplex broaches. In such systems, a portion of the oil is "short-circuited" between the reversible pump and the cylinders so

Fig. 4—Pump parts after 1000-hour run with double-inhibited oil C, Fig. 5. Note freedom from deposits and smooth condition of surfaces subjected to wear



that it is continually affected by the frictional heat generated by passage through restricted parts of the system. Under these conditions, the trapped oil tends to oxidize rapidly and high chemical stability is more important than under less severe operating conditions.

Another cause of accelerated oxidation that involves high temperatures is the compression of free air in the system. The entrainment of air and the variation of the amount of air held in solution with changes in pressure have been referred to previously. A considerable part of the air in solution at high pressure drops out as free air when pressure is released, forming relatively large bubbles. Unless these bubbles break out of the oil before again entering a pressure phase they are rapidly recompressed.

Compression of air from atmospheric to 200 psi, for instance, results in an adiabatic flash temperature of 672 F; similarly, a pressure of 1000 psi means a flash temperature of 1300 F. It should be understood that these are closely localized temperatures and that the quantity of heat in each bubble is extremely small so that there may be no appreciable effect on the bulk temperature of the oil. Nevertheless, the high local heat tends to accelerate oxidation of the surrounding oil. The effects of such influences are cumulative, and if continued over a sufficient period of time, will contribute materially to serious oxidation of the entire batch of oil.

Metallic Contact Speeds Oil Breakdown

Another source of local hot spots is the metallic friction that may occur in various parts of the system. It was previously stated that with a suitable petroleum oil in service, wear is not, generally speaking, a major problem of hydraulic-system operation. However, any metallic contact and resultant wear is accompanied by generation of heat. Partial metallic contact is inevitable, particularly under high pressures. It occurs at high rubbing speeds in vane and gear-type pumps, and on close-clearance reciprocating parts of piston pumps, spool-type valves and cylinders. Obviously the more often a given drop of oil passes through such hot zones the sooner it becomes badly oxidized, which indicates the importance of rate of circulation as a factor in hydraulic oil oxidation.

Oxidation Promoters: This is a term coined to identify certain types of impurities which act somewhat like catalysts in accelerating oxidation. Among the common sources of oxidation promoters are atmospheric dust and dirt which enter the reservoirs through breathers or open covers. Dirty containers used for storing or transferring oil to machines are frequent sources of contamination. Rust and some types of paint particles, which may drop into the reservoir or be washed from other ferrous parts of the system, have a strong catalyzing effect. Wear particles also act as catalysts—another interesting example of the cumulative effect of small local influences on oil oxidation.

Oxidizing effect is most pronounced where small oil volume, frequent cycling, continuous flow through relief valves and other conditions promoting turbulence tend to keep the particles in suspension and dispersed throughout the oil. These conditions promote maximum surface exposure of oxidation promoters to the oil and are, therefore, most effective in catalyzing oil oxidation. When high temperatures and accumulation of oxidation promoters occur concurrently in a system, only an oil of highest chemical stability will serve for extended periods without excessive oxidation.

Demulsibility: Water may be present in hydraulic systems due to leaks from water coolers but accumulates more commonly because of condensation of atmospheric moisture. Unless the water separates readily from the oil, it is entrained and carried through the system where churning by the pump, and the homogenizing effect of control and operating components, tends to promote the intimate mixture of oil and water called an emulsion. The water content in an emulsion has several bad effects: It reduces the lubricating effect of the fluid medium;

Fig. 5—Oxidation-inhibitor life of various oils with additives as indicated by results of ASTM oxidaton test

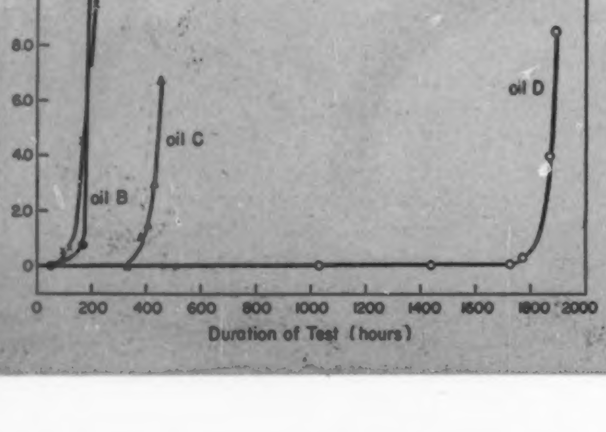
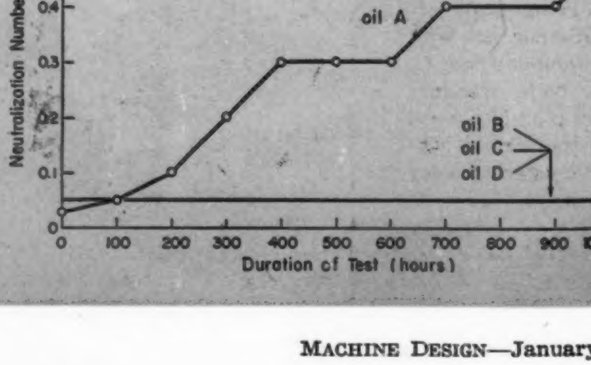


Fig. 6—Results of the Vickers circulation test, 1000 hours under simulated service conditions on same oils as used in ASTM test. These results are not sufficient to indicate the superiority of one additive type oil over another as to potential service life



it promotes rusting; and accelerates oxidation. Persistent emulsions eventually develop into slimy, sticky substances which foul pumps, controls and cylinders, deposit in oil lines and passages, and eventually make it impossible to operate the machines. In order to prevent these results, it is essential that a hydraulic oil have the ability to separate quickly and completely from water. Most reasonably well-refined oils have this ability when new. However, only those oils having exceptionally high chemical stability retain good demulsibility over long service periods.

Effect of Rust: Since most parts of the system are covered with oil as soon as the machine goes into operation, rust formation during operation is probably not the major factor of the rust problem.

Condensation of moisture on unprotected metal surfaces during shipment or storage, or machine down-time, is a logical cause of the heavy rust deposits which cause serious troubles. The surface layers of finely divided rust are picked up by the oil and carried through the system. In this form rust acts much like sludge, which it resembles, in forming deposits in pumps, control valves and cylinders to interfere with the functioning of these parts. Obstruction of filters may also result. Rust in any form is highly effective in promoting oxidation, but particularly when finely divided. Larger rust particles carried into the system can cause severe scoring of pump parts or other moving parts of the system. All of these indicate the need for hydraulic oils having rust inhibiting properties.

Resistance to Foaming: The phenomenon of foaming in connection with petroleum oils has been subjected to much study and research, which has established that there is little variation in foaming tendencies between unused straight mineral hydraulic oils of the same viscosity. However, changes in characteristics due to oxidation, the entrainment of moisture of solid impurities, contamination by cutting fluids, grease, or heavy lubricating oils will all increase tendencies toward severe persistent foaming. Improperly selected additives may also aggravate foaming tendencies. Defoamants may be used to provide some reduction in the foaming tendencies of an oil but the better remedies are to use oils of exceptionally high chemical stability, keeping them clean and keeping air in the systems to a practical minimum.

Additives Improve Oil Performance

High-quality straight-mineral turbine-type oils were found entirely satisfactory for most of the conditions encountered in hydraulic systems prior to the war. In many plants, however, extremely severe conditions imposed by continuous full-capacity production, with insufficient opportunity or manpower for proper maintenance were responsible for oil deterioration, rusting, and other troubles to an extent never before experienced. This led to a demand for better resistance to oxidation and rust formation which was met by the development and use of various additive materials.

Additives have three principal functions: To inhibit oxidation, to minimize rusting and to reduce wear. Following is a brief general explanation as to how these effects are secured.

Oxidation Inhibitors: As previously discussed, oil oxidation is due to combination of oxygen with unstable or "unsaturated" hydrocarbons in the oil. Under sufficiently severe and prolonged combinations of high temperature, severe shear stresses and the accelerating influences of catalysts, even the more stable molecules of highly refined hydraulic oils decompose into less stable forms. In any event oxidation may have two possibly harmful results. One is the formation of acidic materials which, in the presence of moisture and through hydrolysis, may attack and corrode metal surfaces; the other is the formation of large molecules, which first tend to increase oil viscosity and then proceed progressively to form increasingly larger molecules until thin, lacquer-like coatings or heavy, sticky sludges result.

The acidic products of oil decomposition make it possible to determine the extent and check the progress of oxidation by recording the milligrams of an alkali (KOH) required to neutralize the acids in one gram of oil. Results of this test are recorded as "neutralization number." Oxidation inhibitors are considered as functioning, most probably, by interruping the chain reaction of oxidation, thus preventing oxidation from progressing beyond the primary stage in which the oxy-materials are harmless. Some types of inhibitors tend to deactivate or "poison" catalysts, thereby minimizing oxidation.

Inhibitors Selected Carefully

There are two general classes of oxidation-inhibited oils. One type is made by adding an inhibitor to an inherently unstable mineral oil selected because of good "inhibitor response", i.e., the ability to combine readily with inhibitors so as to secure enhanced resistance to oxidation. If an effective inhibitor is used, such an oil may show up very well in laboratory tests and perform well in actual service during an initial period. However, since the inhibitor functions by chemical activity, it becomes depleted after a period varying with the nature of the inhibitor and the severity of oxidizing influences. This is referred to as the induction period, the time required to develop measurable increase of neutralization number in an oil. When the induction period is over, oxidation proceeds rapidly with an oil of this type due to lack of chemical stability in the base oil.

A better type of oxidation inhibited oil is a combination of an exceptionally stable mineral oil and carefully selected inhibitors. Such mineral oil must also have good inhibitor response in addition to high chemical stability. Additives are then selected on the basis of compatibility and activity.

Among the best known tests for determining neutralization number and induction time is the ASTM Oxidation Test. The results secured with four oils are shown, *Fig*. 5. Oil *A* is a top-quality straight-mineral hydraulic oil. Such an oil has been run almost continuously for many thousand hours in hydraulic systems, retaining neutralization numbers below 1.0. Yet, the ASTM Test is so highly catalyzed that this

same oil attains the prescribed neutralization number of 2 in approximately 150 hours, and increases rapidly to 12.5 in about 245 hours. This indicates that the natural inhibitors, even in a top-quality straight-mineral oil, are incapable of resisting the severe oxidizing influences of this test.

Oil *B* is an ineffective combination of a base oil with an inhibitor. The indications are that the inhibitor had a very brief induction period (time required to develop measurable acidity in an oil) and the combination of this inhibitor and a base oil gave practically no greater oxidation resistance than straight mineral oil *A*.

Oil *C* is a somewhat better combination of base oil and inhibitor. Here the induction period is clearly indicated as about 330 hours, followed by the rapid oxidation of the base oil, which it will be noted, is a typical effect of these highly catalyzed test conditions. Oil *D* is indicated as having an inhibitor that imparts high resistance to oxidation under severe catalyzing influences.

As an indication of the fallacy of depending on conventional laboratory tests for determining probable oil performance under service conditions, results of a test run under exceedingly severe simulated service conditions may be considered, ***Fig.*** 6. The standard conditions of this test call for continuously operating a single-stage vane-type pump with constant-volume discharge for 1,000 hours. The system includes a relief valve and contains only 3 gallons of oil; pressure is held at 1,000 psi and temperature is kept constant at 175 F.

Simulated Testing Requires 2500 Hours

The oils are the same as in the ASTM Oxidation Test shown in ***Fig.*** 5. The much milder oxidizing influence of the simulated service test are indicated by the fact that 1,000 hours was required to develop a neutralization number of 0.5 with Oil *A*, and that all three inhibited oils withstood this test at least 1,000 hours without apparent increase in neutralization number. Obviously, unless extended well beyond that duration, this test serves only to indicate that any oxidation-inhibited oil is superior to a high-quality straight-mineral oil in resisting oxidation under a certain set of predetermined, closely controlled, severe simulated service factors. Recent tests indicate that continued service under the stated test conditions must be continued beyond 2500 hours before the actual superiority of one inhibited oil over another becomes apparent.

Rust Inhibitors: Prior to the general trend toward oxidation-inhibited oils to meet the severe operating conditions of the war years, properly refined straight-mineral oils gave adequate protection against rust formation in most hydraulic systems. This was due to the formation of certain oxidation products that coated the metal surfaces and prevented contact with water. The use of oxidation inhibitors prevented the formation of these natural rust-inhibitors, therefore, it was necessary to add materials with rust-inhibiting properties, resulting in so-called double-inhibited oils. Rust-inhibiting additives used in hydraulic oils are highly polar-type materials. Because of their surface activity, they form adsorbed films on metal parts that effectively prevent moisture from contacting the metal surfaces.

High Film Strength Additive: Increased film strength may be secured by special high film strength additives or by some types of oxidation and rust inhibitors. The effectiveness of such additives is indicated by laboratory wear tests that indicate loss of weight of pump parts when operated under wear-promoting conditions over a given period. One such test showed the following comparative loss in weight for two oils of the same viscosity:

Additive Oil	0.03%
Straight Mineral Oil	1.50%

This is borne out by ***Figs.*** 2, 3, and 4, which clearly show the condition of parts after runs on each type of oil.

Careful Design Extends Oil Life

To this point, the discussion has been confined to hydraulic oils, their requirements, characteristics, and differences as to quality. However, since the best hydraulic oil obtainable cannot overcome unfavorable operating conditions, observations as to design weaknesses of hydraulic systems from the viewpoint of lubrication engineers should be discussed in order to promote most effective hydraulic system design.

Oil Capacity: One common fault is small capacity of oil reservoirs. Ample oil volume is required to allow sufficient "rest periods" in the reservoir to insure effective heat dissipation and settling out of impurities. When constant-volume pumps are used, a reservoir capacity of at least twice the pump capacity in gallons per minute is desirable.

Separation of Impurities: Hydraulic reservoirs usually have flat bottoms. Even a slight slope in a tank bottom helps separation by concentrating impurities, and makes draining and flushing easier. On large systems where heavy water contamination is a problem, a water leg will be helpful.

Baffles: Another common fault of tank design is the omission of baffles, which are required to minimize turbulence. Sometimes the oil return is so close to the pump suction that the oil is in continual turbulence and has no opportunity to drop out impurities.

Oil Filters: A continuous full-flow filter capable of removing microscopic particles is desirable to insure protection of rubbing surfaces against abrasion or scoring. By-pass filters are available for installation at suitable points in the system to give additional protection against entry of solids into valves, cylinders, or fluid motors. These, of course, do not prevent accumulation of impurities in the system reservoir.

An important precaution pertaining to filtration is that only cellulose packs, waste-packed or edge-type filters should be used with inhibited oils. Activ-

(*Continued on Page* 154)

Shaft Speeds and Vibrations

. . . measured with capacitance pickup

By Alvin B. Kaufman
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USE of capacitance type vibration pickups and revolution counters has been considerably delayed due to the lack of a simple and efficient "capacity transducer." Such an instrument is required to measure and change small variations of capacity into relatively large variations of voltage or current for indicating rotational speeds or vibrations directly on a meter. Use of a capacitance type rpm pickup permits the measurement of shaft speeds without attaching anything to the shaft or in any way placing a load upon it. This method of determining shaft speed was used by the author to measure aircraft cabin supercharger impeller speeds in excess of 30,000 rpm. Photoelectric and stroboscopic methods tried previously were unsatisfactory for production and test use because of interference by fluorescent lights and mechanical installation problems. Mechanical problems also made the use of magnetic pickups unsuitable. Vibration pickups of the capacitance type allow vibration measurements to less than ½ cps and to any amplitude and acceleration required.

Three units are required to indicate vibration or rpm; the pickup, a capacity transducer and an electronic frequency meter. The pickup for rpm indication consists of one or more insulated vanes, located adjacent to the rotating shaft or blades so as to vary the capacity to ground of the pickup vanes when the shaft is rotated.

Vibration capacitance pickups are not new. They employ, in general, two elements the capacity of which varies with the rate of vibration frequency and the capacity change of which may or may not be relative to the vibration amplitude. Static and dynamic capacity of such a pickup is supplied to the capacity transducer herein discussed, which in turn supplies an alternating voltage, the frequency of which is proportional to shaft rpm or vibration, depending upon the application. This alternating voltage may be supplied to an electronic frequency meter whose scale may be read directly in vibration service, or translated into terms of rpm for tachometer use.

A capacity transducer must be simple, relatively free from drift, and of reasonable accuracy for the application. The unit described here, shown in *Fig.* 1 with the author, has these features and eliminates some of the disadvantages possessed by the standard bridge or FM discriminator circuits now in use. The sensitivity of the transducer unit is such that with a single amplifier tube, outputs of three rms volts are

Fig. 1—The author adjusts a capacity transducer equipped with a single-vane impeller pickup. The electronic frequency meter used to read rpm or vibration frequency is not shown in the photograph

available for a one or two micro-microfarad capacity change.

The capacity transducer unit changes the variations in pickup capacity into a useful audio frequency voltage suitable for application to an electronic frequency meter, tachometer, or other indicating or listening device. This unit consists of the familar "capacity relay" or radio frequency oscillator, a detector, and proper amplification. The unit shown in *Fig.* 1 was used for measuring the rpm of the DC-6 supercharger impeller seen in *Fig.* 2.

Cable Selection Important

Where this unit is used for high rpm measurements or vibration frequencies in excess of several thousand cycles it is necessary to use RG 7/U cable, or any other low-capacity cable, to connect the transducer to the electronic frequency meter. Inasmuch as the pickup unit works on a capacity change it is necessary that its connecting cable be low in capacity and that it have little capacity change when moved or vibrated. If the total capacity of the connecting cable were excessively high, variation in pickup capacity might fall to such a low percentage of the total input capacity as to seriously affect the signal-to-noise ratio. A suitable cable is RG 8/U or 7/U with appropriate coaxial plugs and receptacles. The pickup must fasten securely to the unit under test to minimize capacity variation with vibration. Variation of pickup capacity should be at least five to ten per cent of the total input capacity.

When measuring rpm it is desirable to know the expected output frequency, from which the shaft rpm can readily be calculated as follows: Output frequency = number of blades or projections on the shaft times the rpm divided by 60. As an example, consider a four-bladed fan which rotates at 1800 rpm; $4 \times 1800 = 7200$; $7200/60 = 120$ cycles per second. The rpm may be read directly on an electronic tachometer where one cycle is produced per revolution at the pickup. Where many-bladed devices are used, *Fig.* 2, it is preferable to use an electronic frequency meter. In this case, rpm may be read by using the calculation: Rpm = (frequency meter reading $\times$ 60)/(number of blades). This calculation may be reduced to chart form for any particular installation.

Vane-to-Blade Clearance Not Critical

A one-vane pickup, *Fig.* 1, may be used for most rpm measurements. The size and shape of the vane are not critical, but are chosen so that the projection or blade on the shaft is under the pickup plate or vane for less than one-half the width of the pickup vane before the next blade passes under the vane. This gives roughly a 1 to 1 high-to-low capacity cycle. The transducer converts this into an alternating output wave. Standard electronic tachometers used with the transducer require an "on to off" time or alternation of input voltage of preferably 1 to 1 but not to exceed 4 to 1 for highly accurate indications. The space between pickup vane and rotating elements depends upon transducer gain and available variation of pickup capacity, but is not critical. Spacings up to several inches are possible. Restrictions of air flow, as in fan or turbine measurements, may be limited to a low value by proper design of the pickup.

It should be noted that the number of pickup vanes

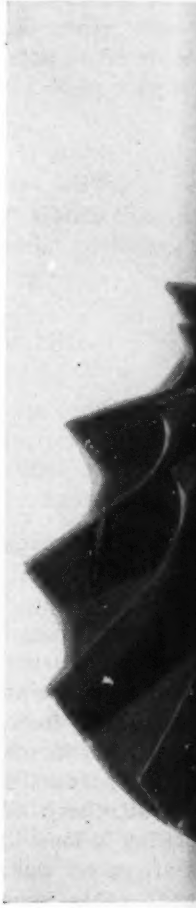


Fig. 2—Right—Supercharger impeller from DC-6 cabin pressurizing system. Four-vane pickup was used to measure impeller speeds of as high as 30,000 rpm

Fig. 3—Below — Four-vane pickup. Vanes are machined from one piece of dural and insulated from frame

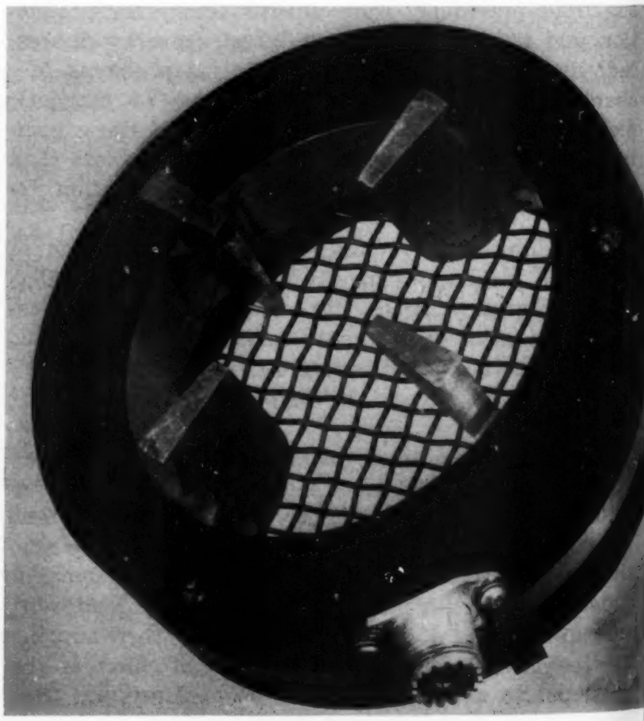
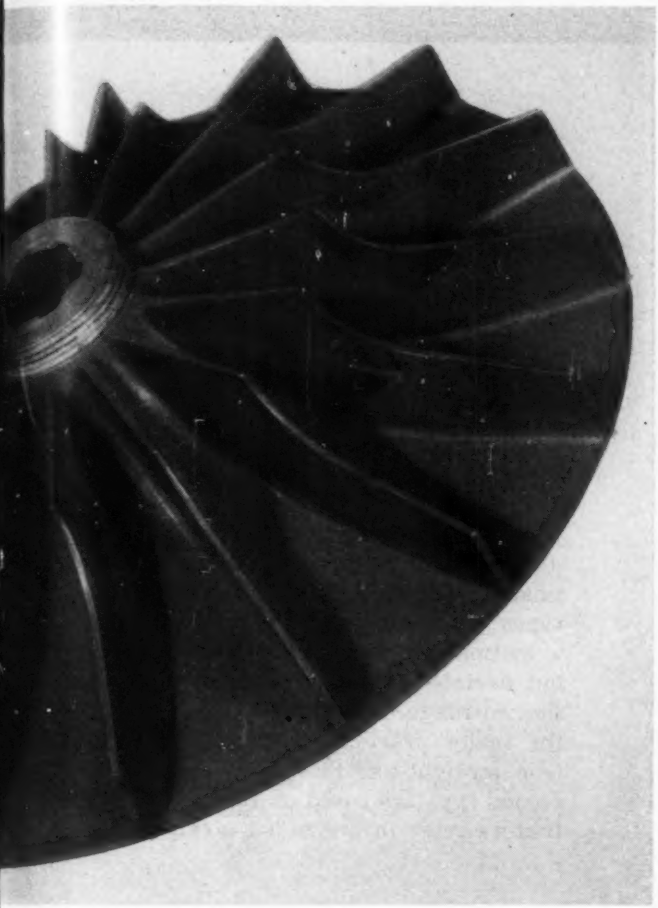
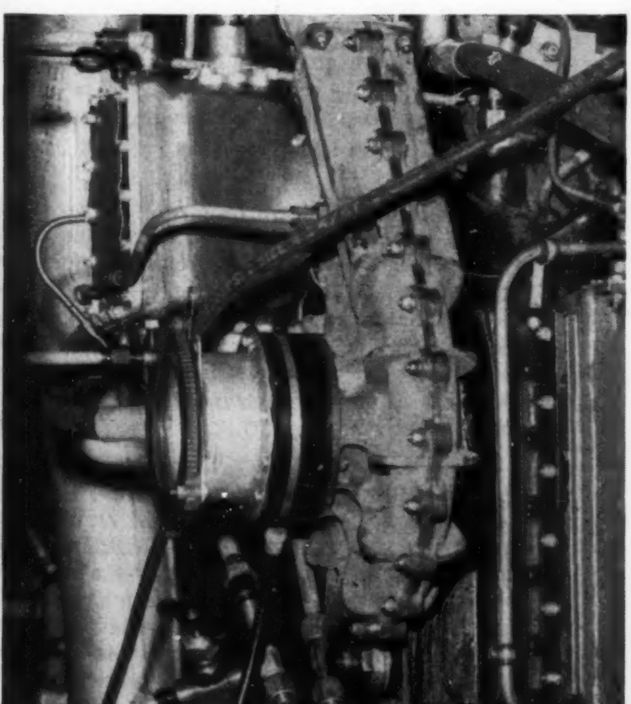


Fig. 4—Below, right—Impeller test setup showing transducer pickup mounted between air intake and supercharger scroll

does not change the output frequency; rather, the number of projections or blades on the rotating shaft varies the frequency. This is because all of these pickup vanes are electrically connected together and are spaced so as to approach and pass the shaft blades synchronously. Thus, all the pickups together produce no more capacity pulses than one vane; just a change in amplitude.





A four-vane pickup, *Fig.* 3, was used to measure the airplane supercharger impeller rpm in the test setup shown in *Fig.* 4. The reason for this was not the increase of capacity change and resulting greater transducer output. Rather, under extremes of high vibration or weaving of the rotating blades at very high speeds, a high degree of "hash" may be produced which will prevent operation of the system. These stray variations may be corrected with the use of a three or four-element pickup which will balance out the weaving capacity changes. As one blade moves closer to a vane, its increase in capacity is balanced by the other blade (180 degrees away) moving away from a vane. With the four-vane pickup the 16-bladed impeller on the Douglas DC-6 supercharger, operating at 30,000 rpm, caused an output frequency of 8000 cycles to be delivered from the transducer to the frequency meter.

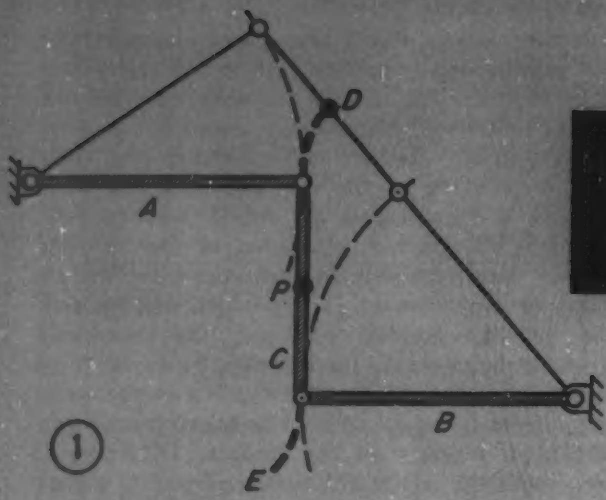
Testing Aircraft Wing Characteristics

Conventional capacity vibration and acceleration pickups will not operate at the low frequencies within which the transducer is capable of functioning. For these measurements, the transducer amplifier is direct coupled and the pickup is fabricated of two parts. As an example, it is desired to test the deflection and vibration characteristics of an aircraft wing before flight. The wing is mechanically shaken, with any of the available commercial vibrators, throughout the expected range of aircraft vibration frequencies. In this case the metal wing is connected to the transducer and serves as the "cold" or ground plate of the pickup capacity. The "hot" plate of the capacity pickup is insulated and supported from the ground next to the wing at an appropriate distance. Here the wing may move at any rate and its capacity changes to the insulated plate may be recorded in terms of vibration frequency, amplitude or acceleration.

The simplicity and high gain inherent in this type of transducer make it applicable to any problem where it is desirable to change a capacity variation into a useful control or measuring voltage.

Oscilloscope Records Four Channels

A CATHODE-RAY oscilloscope for research work that can indicate as many as four variables on a single 5-inch, flat-face tube has been developed by Electronic Tube Corp. The four-channel oscilloscope was designed primarily for use with a continuous film type of recording camera. Separate focus, intensity and positioning controls are provided in each of the channels, and positioning in both horizontal and vertical directions is possible. Because the common linear sweep as provided by the film drive is in the vertical or *Y* direction, the four independent signals are applied to deflect along the *X* axis. Horizontal deflection can be measured in terms of volts by a direct-reading calibrator which supplies a square wave of known adjustable amplitude to the signal amplifier.



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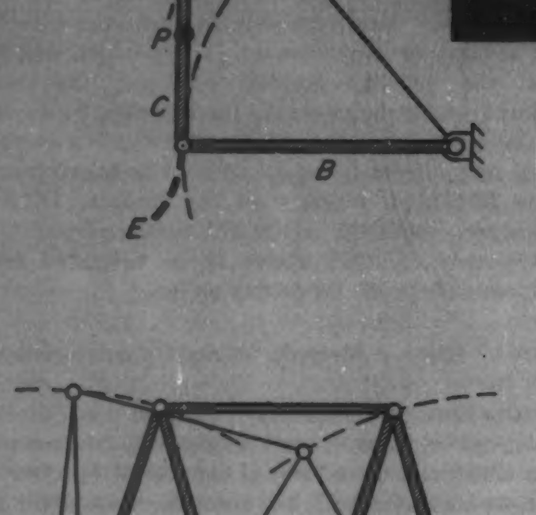

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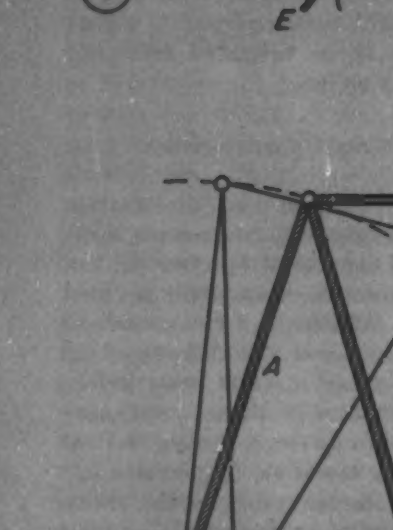

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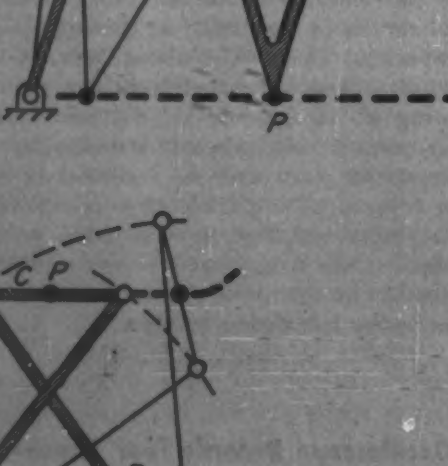


4

Straight-Lin

By H. G. Conway

FEW mechanisms are more intriguing to study and develop than complex linkages designed to produce specific types of motion. The linkages presented in this pictorial, all yielding straight-line motion, are classics in their field. Close study of them will bring to light basic principles which might well be applied to the solution of a variety of problems in machine design.

Sometimes referred to as parallel-motion linkages, straight-line linkages are of two types: approximate and exact. The first give a motion which is arc or S-shaped in fact, but deviates from a true straight line over a limited range of motions by so little as to be, for many practical purposes, equivalent to true straight-line motion. Nearly all of the second type are based on the geometrical fact that a circle's inversion is a straight line.

Fig. 1—WATT, 1784, (approximate): Two swinging links, *A* and *B*, are connected by a link, *C*, the mid-point, *P*, tracing an S-curve, the central part of which closely approximates a straight line. Deviation at *D* and *E* is appreciable and thereafter is considerable.

Fig. 2—ROBERTS, (approximate): Two equal links, *A* and *B*, are hinged at a distance apart of twice the base of the isosceles triangle central member *C*. Point *P* traces a curve which passes through the pivots and the mid-point, deviating from a straight line by a very small amount between the pivots, but appreciably outside them.

Fig. 3—TCHEBICHEFF, 1873, (approximate): *L* must be four-fifths of *A* or *B*, *C* must be one-half of *L*. The mid-point, *P*, of *C*, traces a curve very nearly a straight line over a distance approximately equal to *L*, deviating thereafter. The line passes through the points vertically above the pivots and at the same height as the central position of *P*. Other close ratios of link lengths give similar results.

Linkages

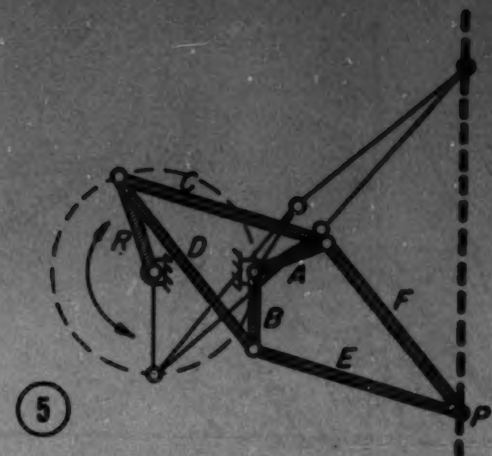
Fig. 4—PEAUCELLIER'S CELL, 1864, (exact): Two links, *A* and *B*, are hinged together at the pivot, *Q*, and at their other ends to the diamond linkage *CDEF*. The apex between *C* and *D* is hinged to a radius member, *R*, describing a circle passing through *Q*. The motion of *P* is exactly a straight line between the limits of *A* being in line with *C* and *F*. If the pivot, *Q*, is moved to the left, *P* describes a circle of large radius, concave towards *Q*. If *Q* is moved to the right, *P* describes a circle of large radius, convex towards *Q*.

Fig. 5—INVERTED PEAUCELLIER, (exact): A variation of the original linkage where the members *C, D, E,* and *F*, are longer than *A* and *B*. The motion again is exact straight line between the limits of the links *A, C* and *F* being in line.

Fig. 6—EXTENDED PEAUCELLIER, (exact): A variation of the original linkage where the members *C, D, E,* and *F* are connected by extended links A^1, A^2 and B^1, B^2. The motion is exact over the comparatively long distance limited by the links A^1 and *C* being overfolded and A^2 and *E* being extended and in line.

Fig. 7—HART'S, 1874, (exact): In this, a contra-parallelogram, *ABCD*, is pivoted at the midpoint of *A*, the midpoint of *D* being hinged to a radius member *R*, describing a circle passing through the pivot of *A*. *P*, the mid-point of *B*, describes an exact straight line up to the points where members *A, B, C* and *D* are in line.

Fig. 8—SYLVESTER-KEMPE, 1875, (exact): On this, three bell-crank members and one straight link are hinged together at their ends. Motion of *P* is truly straight within the limits shown in the two positions (a) and (b). The motion is most unusual and is best understood from a model.



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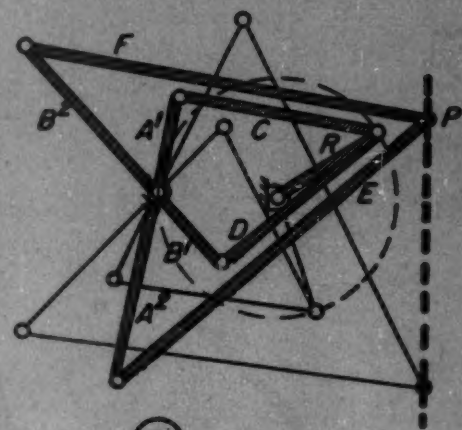

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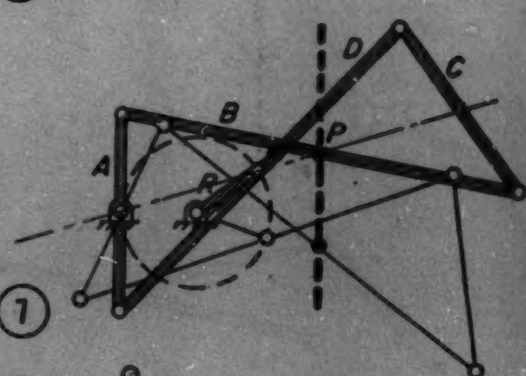

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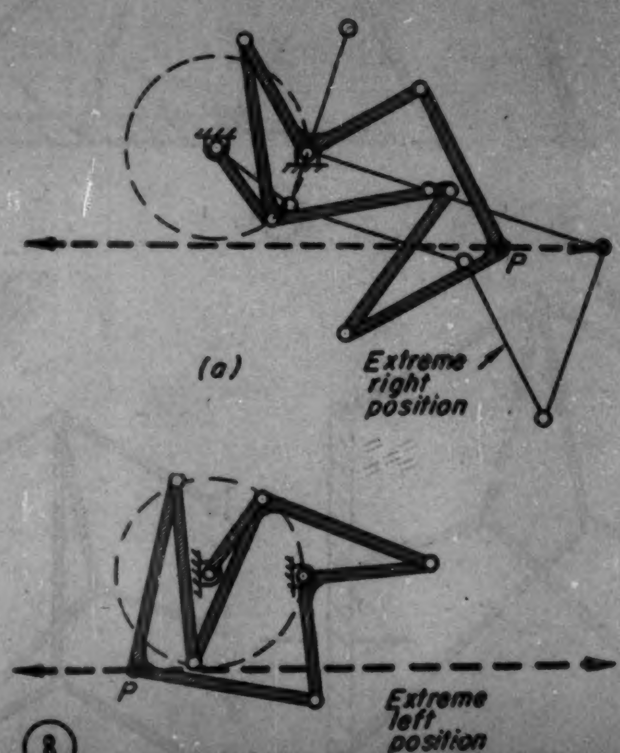

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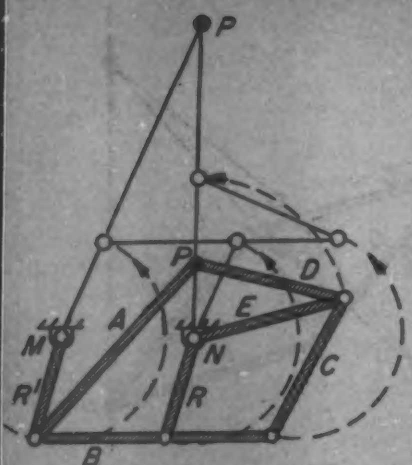

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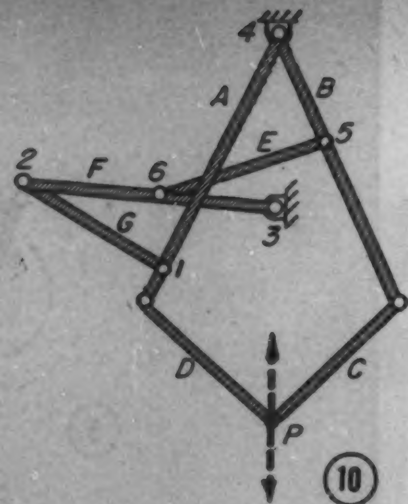

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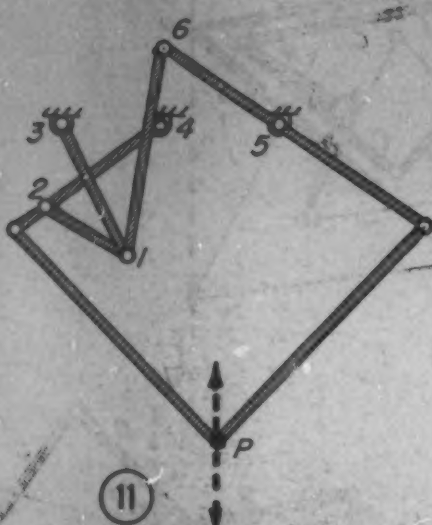

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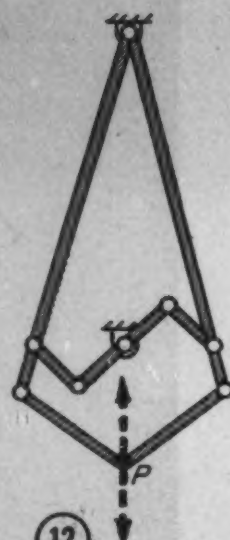

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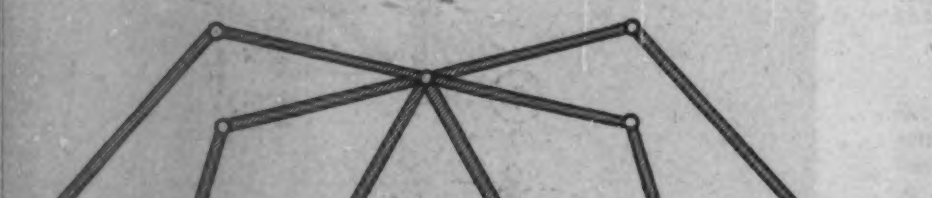

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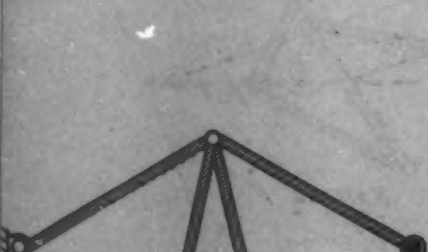

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Fig. 9—KEMPE, 1875, (exact): This has two similar quadrilaterals or diamonds *A, B, C, D,* and *C, E, R,* and part of *B.* ($A{:}D = E{:}R$) *R* is pivoted at *N* and a second radius member, *R'*, at *M*, so that the motion of *B* is always parallel to *MN*. *P* describes a true straight line through *N* of an amplitude $2(E + D) = 4D$.

Fig. 10—KEMPE, (exact): If two links, *A* and *B*, can be arranged to hinge equally but in opposite directions, the two connecting links, *C* and *D*, move the point *P* in a straight line through the hinge point of *A* and *B*. The three auxiliary links, *E, F* and *G*, give the appropriate reversing motion to *A* and *B*. These links form two contra-parallelograms, (1, 2, 3, 4) and 4, 5, 6, 3), with one common side (3,4) and similarity, i.e., (1, 2) : (1, 4) = (4, 5) : (5, 6).

Fig. 11—KEMPE, (exact): This is similar to that of Fig. 10 with the links and the two interconnected contra-parallelograms, (1, 2, 3, 4) and (1, 3, 5, 6), arranged differently. The condition of similarity of the two contra-parallelograms applies.

Fig. 12—SYLVESTER, (exact): This is also similar to that of Fig. 10 but uses a simple reversing linkage to give the contrary motion of the two main links. The same object can be achieved by gearing the two levers together (Cartwright's linkage).

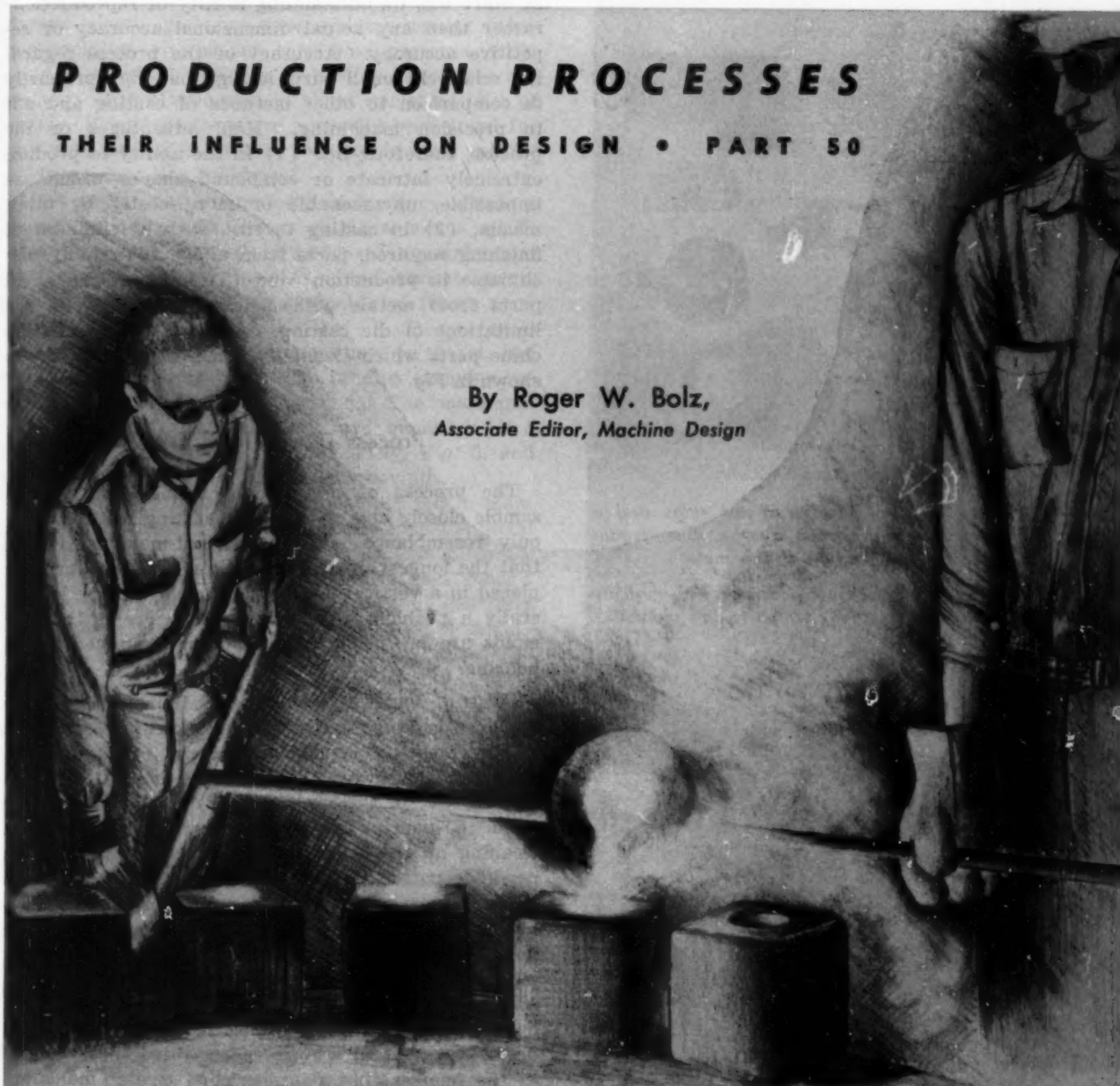
Fig. 13—KEMPE, (exact): The two interconnected pairs of similar quadrilaterals guide the whole link, *A*, along a straight line passing through the linkage pivots.

Fig. 14—KEMPE, (exact): As in Fig. 13, two interconnected pairs of similar quadrilaterals guide the motion of link *A* at right angles to a line passing through the pivots of the linkage, the motion having only this direction.

Fig. 15—KEMPE, (exact): This is a variation of the linkage of Fig. 14, imparting precisely the same motion to link *A*.

PRODUCTION PROCESSES

THEIR INFLUENCE ON DESIGN • PART 50



By Roger W. Bolz,
Associate Editor, Machine Design

Investment Casting

LAST but not least in the group of major casting methods is that best termed as investment casting. An ancient process, it was known and used by the Chinese prior to the year 50 B.C. Benvenuto Cellini used the process to create his celebrated "Perseus with the Head of Medusa". Modern history of investment casting, however, dates from shortly after the turn of the century when Dr. W. H. Taggart interested the dental profession in the advantages of the process in producing metal dentures.

Developments and improvements in technique which followed led to widespread usage of the process by the jewelry industry in the middle thirties. Value of the process for solving many problems in the production of intricate machine parts first gained recognition during World War II and from the resulting industrial experience, *Fig.* 1, the investment casting process has become one of the present basic production methods.

Contrary to popular opinion, the major factor which influenced application of the process from

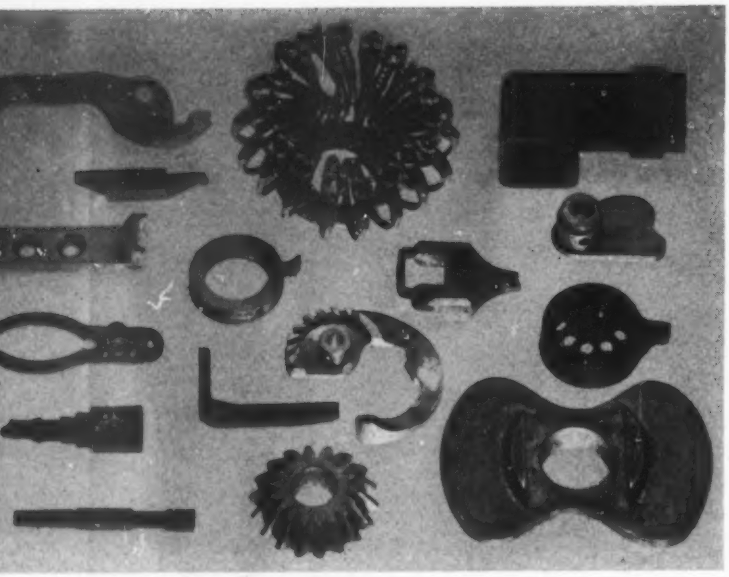


Fig. 1—Above—A "tree" of investment-cast rings and a variety of machine parts produced by the same means using centrifugal feeding of the metal

Fig. 2—Below—Group of investment castings with intricate shapes typical of those ideally suited to this method

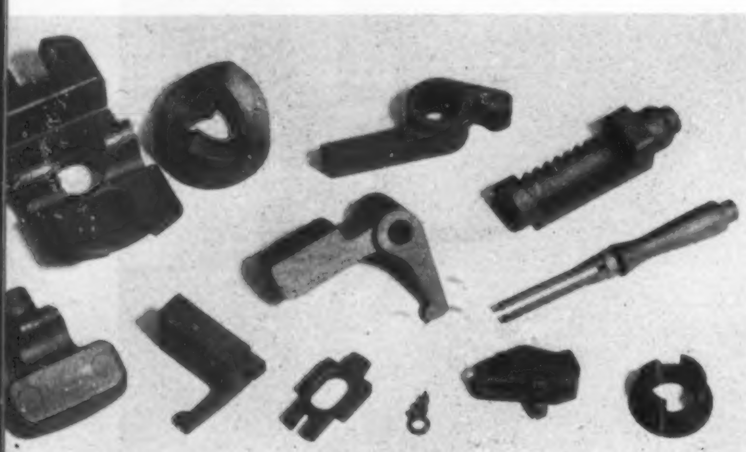


Fig. 3—Master pattern is embedded in plaster of paris in one-half of the master-mold flask to pour the soft-metal mold for the opposite half

the start was its outstanding fidelity of reproduction rather than any actual dimensional accuracy or repetitive accuracy. Accuracy of the process regarding relatively small parts is high but high primarily in comparison to other methods of casting and not to precision machining. Main advantages of the process, therefore, lie: (1) In the ability to produce extremely intricate or compound shapes which are impossible, unreasonable or more costly by other means; (2) in casting to size, with a minimum of finishing required, parts from alloys not readily machinable in production; and (3) in producing small parts from metals with melting points beyond the limitations of die casting. A group of typical machine parts which exemplify some of these factors is shown in ***Fig.*** 2.

Process Differs from Others

The process of investment casting does not resemble closely any of the other casting methods. The only resemblance is to permanent-mold casting in that the longest dimension of a cast part is normally placed in a vertical position and mold venting is generally a problem. The major difference is that the molds are one-piece and cannot be examined before pouring.

Casting Process: In general the process of investment casting consists of the following basic steps: (1) Forming the expendable pattern, (2) gating and assembling patterns, (3) investing the patterns, (4) melting out of patterns and drying the investment mold, (5) pouring the castings, and (6) removing and cleaning up the castings. Production of accurate expendable patterns is all-important in order to obtain the necessary results. Master molds for production of patterns in quantity are made of rubber, low melting point metals or steel. Rubber and soft-metal master molds are made by vulcanizing or casting against a master pattern of the part desired, which is generally made with an allowance for shrinkages which occur and held to tolerances one-third those desired in the final casting, ***Fig.*** 3. Steel master molds are generally sunk directly from the drawing specifications.

Use of rubber molds is more or less confined to the casting of jewelry. Soft-metal molds are generally found satisfactory for parts without too many complications or close tolerances. They are generally preferred, however, for parts having contours or shapes which are difficult to generate or reproduce, as with turbine blades, etc. Steel molds are most satisfactory for long runs and for parts requiring the maximum in dimensional accuracy or having small cores, thin slots and intricate, involved shapes.

Expendable Patterns: Although the process is sometimes termed the "lost-wax" method, this is a misnomer inasmuch as wax is not the only pattern material utilized. Numerous materials have been employed for patterns but today the materials in common use are waxes, plastic and frozen mercury. Properties of each are different, require different handling and are suited for different applications.

Waxes are probably most commonly employed in

production, being simplest to handle, *Fig.* 4. Polystyrene plastic patterns, produced by regular injection molding, are harder than wax patterns, withstand more abuse and can be held to very close tolerances. The plastic cost is about one-third that of waxes. Plastic patterns are generally best suited for long runs owing to greater expense in steel master molds and injection equipment.

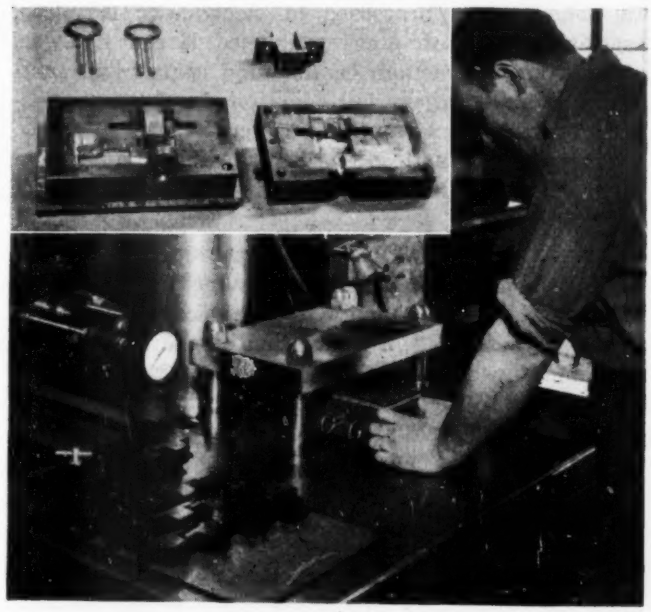
Most recent pattern material is mercury which is poured into dies and frozen at below minus 40 F. Advantages are claimed in improved accuracy and greater size of part possible. More intricate parts may be cast by assembling a number of frozen mercury pattern pieces.

Frozen mercury patterns, like those in waxes, are readily bonded to each other to produce complex assemblies or to sprues for casting. The maximum number of patterns practicable are mounted on a single sprue for casting simultaneously, *Fig.* 5, and minimum size thus permits obvious savings.

Investments: Type of investment utilized in making the final mold for casting depends upon the metal to be cast. That used for nonferrous alloys generally consists of silica with plaster of paris as a binder. Ferrous alloys which react with plaster generally require other investment materials such as a refractory and a suitable binder which also will resist the higher temperatures involved.

In preparing high-temperature investments a primary investing dip coating is given the patterns, *Fig.* 6. Typical is one of sodium silicate, a wetting agent and silica which is dusted over with a somewhat coarser grade of silica. After attaching the flask to the bottom plate on which the patterns are mounted, the secondary investment (typical is one of silica, grog of ground up fire brick, and a mix of alcohol and tetraethylsilicate plus one-half old investment) is poured around the patterns. This is jolted during the short setting period and after hardening the patterns are melted out. Following a final thorough preheat the metal is poured, *Fig.* 7.

With mercury patterns a clay such as kaolin is



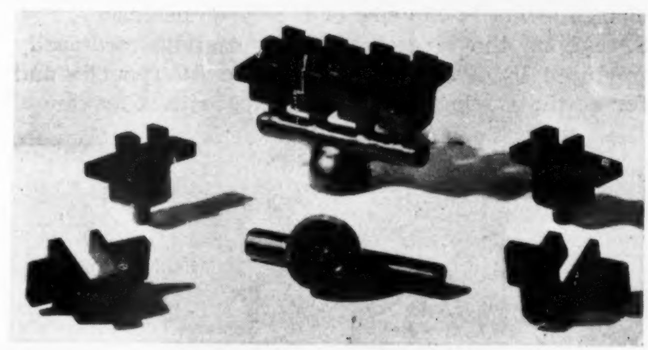


Fig. 4—Right Above—Master soft-metal mold with corepiece and wax patterns being produced on typical injection press

Fig. 5—Right—Sprue and wax patterns and finished group mounted ready for investment

Fig. 6—Below—Dipped and dusted patterns are dehumidified thoroughly before pouring the secondary investment

Fig. 7—Below Right—Pouring a group of investment castings by common static method

often used. This is deposited in layers normally to ⅛-inch in thickness, by dipping, although up to ½-inch has been used. After setting at low temperature the mercury pattern is allowed to melt out or is flushed out with warm mercury. The shell is then fired to produce a hard, permeable ceramic mold which is surrounded with dry sand, preheated and the casting poured.

Casting: Investment castings are poured by static, centrifugal, air pressure, or vacuum methods. High-temperature alloys and small charges (up to about 15 pounds) are processed in electric-arc furnaces, especially where air pressure is used. Larger charges, static pouring, vacuum casting and centrifugal casting are done largely in induction furnaces or gas-fired crucibles. Advantages of centrifugal pouring, as originally practiced in dental work, is gaining favor owing to improved metallurgical character of the cast metal.

Size of Castings: Generally speaking, investment casting is best suited for the production of reasonably small parts. Maximum dimensions and weight of parts are limited in production by the flasks available and furnace capacities. The largest cylindrical mold generally used is 10 inches in diameter by 20 inches long and the largest box flask is 9 by 12 by 15 inches.

Maximum capacity of pressure pouring furnaces available is about 15 pounds. Induction furnaces up to 150 pounds capacity are used for static pouring. Maximum weight of a part or parts poured generally does not exceed 50 to 70 per cent of the furnace charge owing to the size of the sprue necessary, *Fig.* 8. About the largest pressure castings ordinarily produced in quantity range around 3½ pounds and for static pouring about 15 pounds with a maximum dimension of approximately 6 to 8 inches. Parts up to 18 inches long and weighing almost 100 pounds have been poured statically. Smallness of size is practically unlimited—one of the smallest parts produced to date weighs less than 0.002-pound.

Production: Investment casting offers no competitive advantage for large quantities of parts which can be produced in several setups on automatic machinery. Where the quantity is small, however, the lower cost of tooling can often make the casting more economical. One specific example of this nature indicated that the point of equal cost per piece was reached at 5000 pieces. The limit, of course, varies with the particular part design. Production runs may thus range from as few as six pieces to as many as 100,000 pieces. Economical production runs for average machine parts generally range from 500 to 5000 pieces.

Considerations in Design

Design: In general, any relatively small part which would require a fair amount of machining to produce is a possible candidate for investment casting. Where overall cost of parts, machining, fasteners, and assembly can be eliminated by one investment casting requiring but a small amount of finishing, highly economical results can be obtained, *Fig.* 9. Cost savings per piece as high as 95 per cent have been recorded in numerous cases. Many otherwise expensive operations can often be eliminated. Typical of these are precision operations such as drilling, threading, tapping, tapering, grinding, etc., usually necessary for assembly purposes. The casting in *Fig.* 10 eliminated a shrink fit.

The ideal part for this method of production weighs

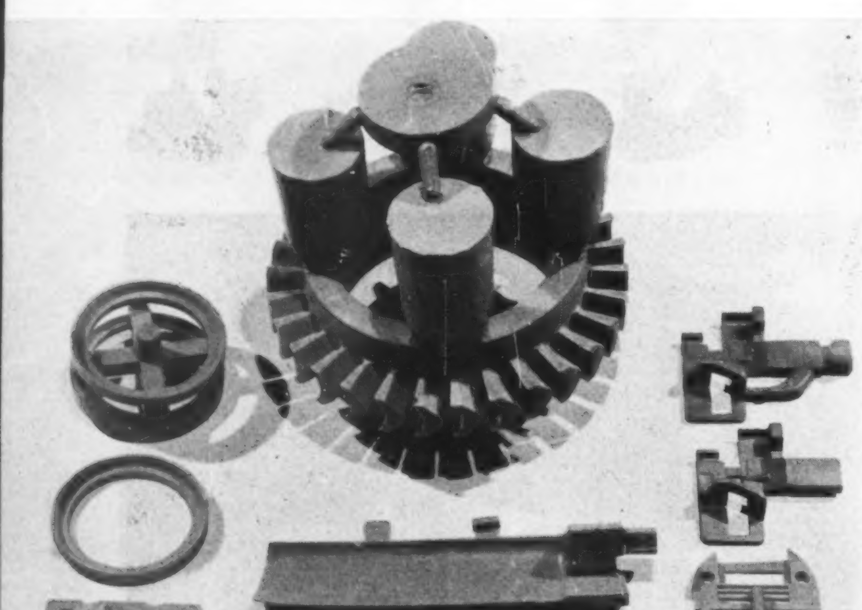
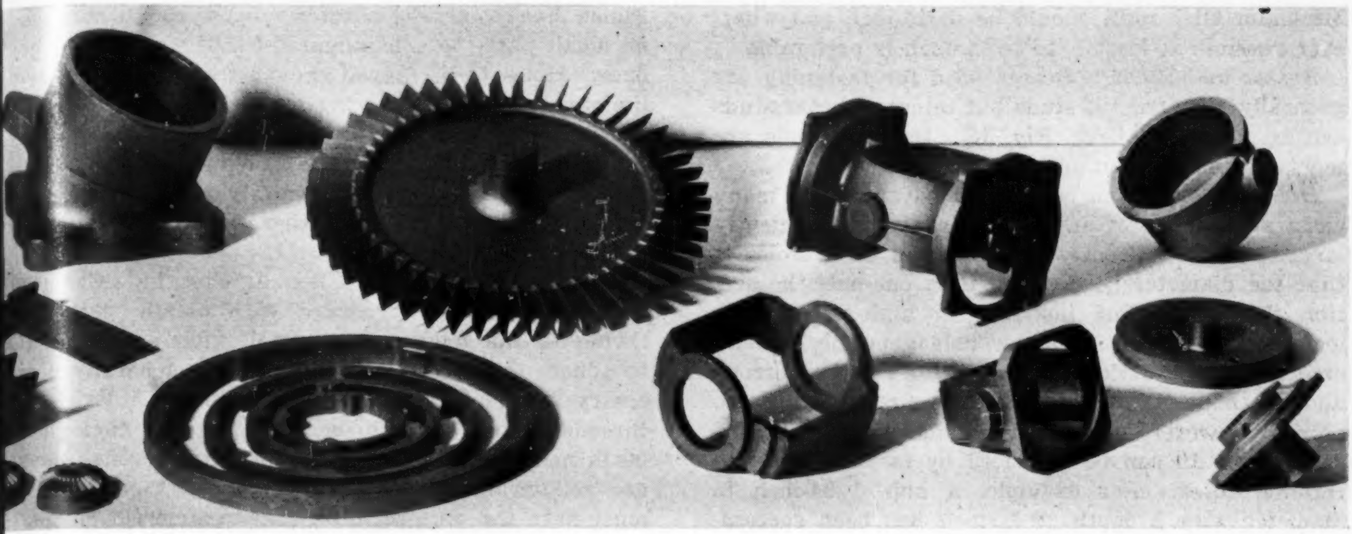


Fig. 8—Left—Gated pattern for a rotor disk showing size of sprues used. Other patterns include a conveyor crank and turbine blades

Fig. 9—Below—Beryllium-copper calculator variable cam with a maximum dimension of about 2 inches as investment cast. Original cam cost $30.00 to assemble from stampings by brazing while casting cost approximately $2.00



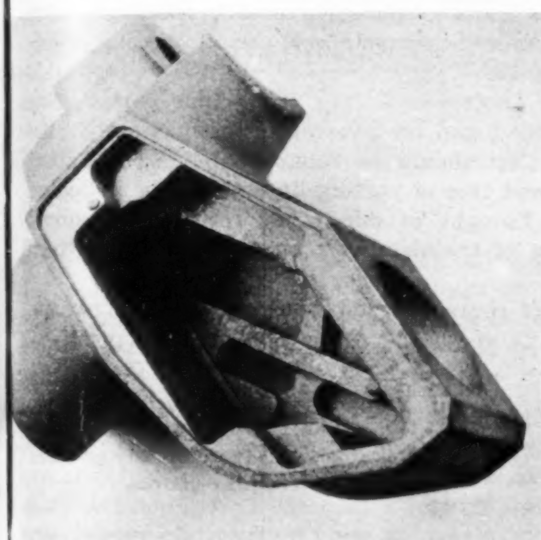
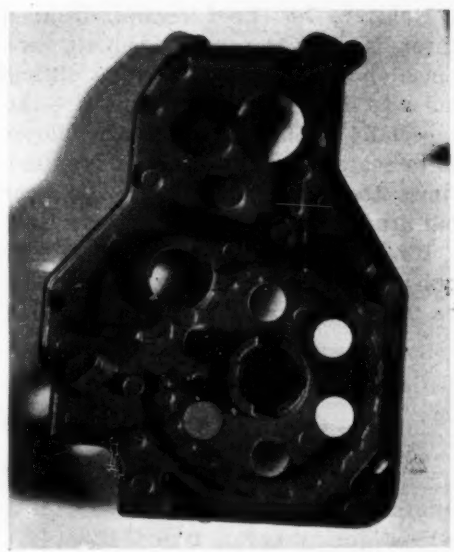


Fig. 10—Left—This binocular housing, cast in magnesium, formerly required a shrink fit assembly of two parts

Fig. 11—Above—Group of typical castings showing widely varying sections and range in size and complexity of design

Fig. 12 — Right — Cast machine side frame showing numerous cast studs and shaped bosses



in the range from approximately one ounce to one pound and has uniform sections within the range of 0.035-inch minimum, to ¼-inch, maximum. Good design for casting in general applies in the case of detail design for investment casting and must be adhered to for good results.

Sections and Thickness: Though the preceding specifications indicate that uniform sections from 0.035 to ¼-inch are ideal, castings with sections from 0.020 to ⅜-inch and occasionally ½-inch are actually produced. Sections as light as 0.015-inch are possible where they are continuations of tapered sections and do not exceed ¼-inch in length. It is unwise to demand, unless absolutely necessary, any section continuously less than 0.025-inch. For large areas, wall thickness should never be less than 0.050-inch. Areas up to approximately 12 square inches may be cast in sections as thin as 0.040-inch, minimum, with reasonably accurate flatness where adequately supported or ribbed. Also, heavy sections should be avoided and ribbing substituted to obtain the desired stiffness. Uniformity is ideal but, of course, varying sections such as in turbine blading are feasible where desired, *Fig.* 11. Wherever sections vary, however, the transition should be gradual and the effects of uneven cooling between sections of varying thickness should be carefully considered.

Edges: Thin edges are difficult to produce in wax without distortion or handling difficulties. The minimum practical edge thickness is generally considered to be 0.012 to 0.015-inch provided there is no abrupt section change and a gradual taper is used. Thinner edges have been cast, i.e., a trailing edge of 0.005-inch on a blade section of 0.015-inch, but are not practical as a rule, especially in the hard alloys.

Edge radii of 0.005-inch can be held at the crests of serrations having a fine pitch and wide angles. In casting serrations some difficulty is encountered with small globules of metal which often form at the roots of sharp or small included angles.

Draft: Investment castings do not require draft on outer walls or cores regardless of the fact that the expendable patterns are formed in permanent type dies. Occasionally, however, it is found advantageous to use a draft of about ½-degree to permit proper removal of the patterns. Those surfaces which should have no draft should be indicated to permit use of draft elsewhere when necessary for production.

Fillets: Generous fillets are advantageous both functionally and in producing a stronger pattern.

Minimum fillet radii should be 0.015-inch and wherever possible at least 1/16 to ⅛-inch is preferable.

Bosses and Studs: Bosses used for fastening are generally preferred to studs but integrally cast studs can be used if desired, *Fig.* 12. Integral studs are more economical than inserted types.

Holes: Coring can be used to advantage in many cases and can be of any size or shape if sufficiently large, *Fig.* 13. General limitation on cored holes is that the diameter be not less than one-half the section thickness. For instance, in high-temperature metals the smallest diameter hole generally considered feasible is 0.050-inch and in the soft nonferrous alloys, about 0.020-inch. Where circumstances so permit, however, smaller holes and holes with l/d ratios over 10 can be produced by means of special ceramic cores. For example, a hole 1/64-inch in diameter with a depth of ¾-inch has been successfully produced.

Holes to be cored should, if at all possible, be through holes to permit adequate core support. Blind holes should be avoided. It is difficult to cast blind holes to depths greater than twice the hole diameter, especially in high-temperature alloys. A good rule is to restrict blind holes to a maximum depth of 1¼ times the diameter. Undercuts, again, are possible but are best avoided for economy.

Machining Allowances: Those surfaces which require closer tolerances than this method provides must be machined and an allowance for this must be made. Finish stock generally allowed ranges from 0.010-inch on small parts to a maximum of 0.040-inch on large ones. Holes to be tapped should always have a preliminary reaming or boring operation to assure best possible threads.

Threads: Barring special unmachinable alloys, size or form of thread, it is generally more economical to cut or grind threads. Either external or internal threads can be produced but the resulting accuracy seldom exceeds that of Class 1, *Fig.* 14. Class 2 threads have been produced with plastic patterns. Owing to the tendency for small globules of metal to adhere to the roots of threads it is generally necessary to clean them up by machining. External threads smaller than ¼-inch and pitches finer than 20 in nonferrous alloys or 14 to 16 in ferrous alloys are seldom feasible. Threads longer than ½-inch must have the pattern pitch compensated for shrinkage. Where close "machining type" tolerances are desired it is not recommended that threads or gears be cast, unless circumstances permit casting tolerances, *Fig.* 15.

Parting Lines and Gating: A visible parting line is generally found on investment castings and in design this fact should be kept in mind. Areas that must be kept free of parting lines should be indicated with the thought in mind that restrictions should permit use of the simplest possible two-piece pattern die.

Gates, of course, must be used to feed the casting

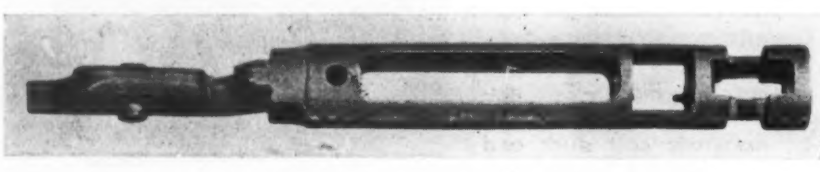
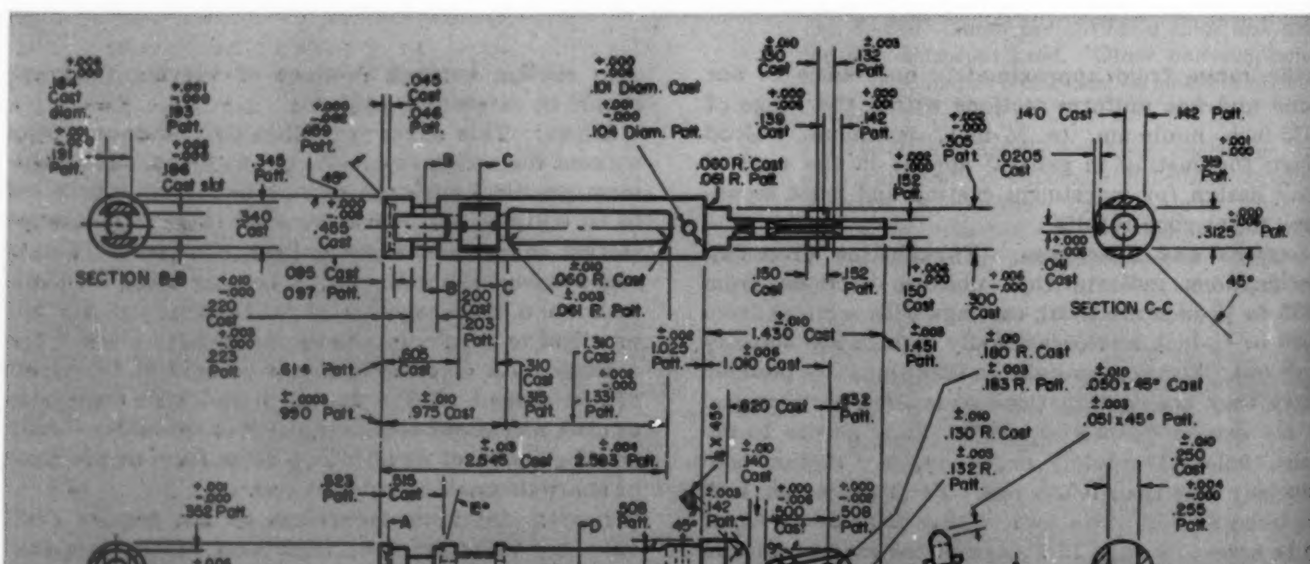
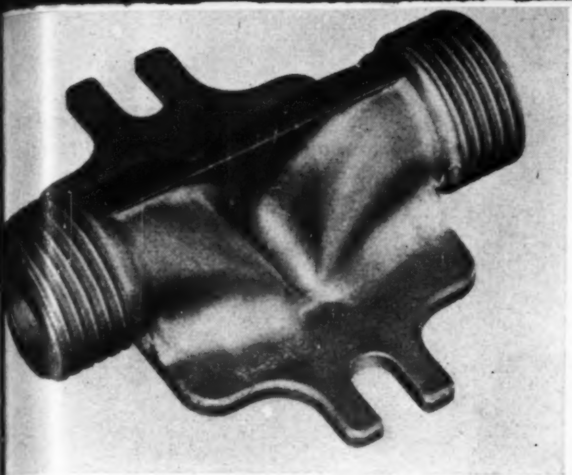


Fig. 13—Aircraft clamp piece used for quick disconnect on engine, cast in 410 stainless steel, showing a variety of holes and complex shapes





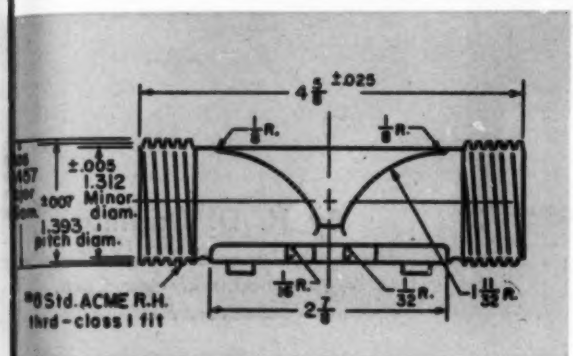


Fig. 14—Left — Cast fitting with Acme threaded end connections used as cast

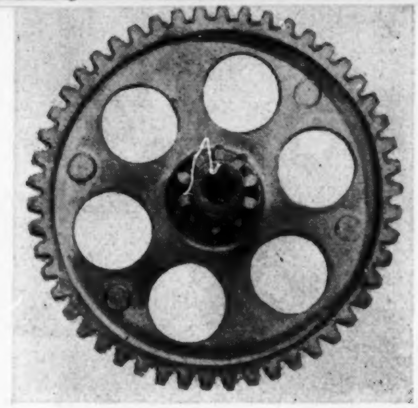


Fig. 15—Top Right—Investment-cast Hastelloy alloy C gears for a steel mill pickling tank drive

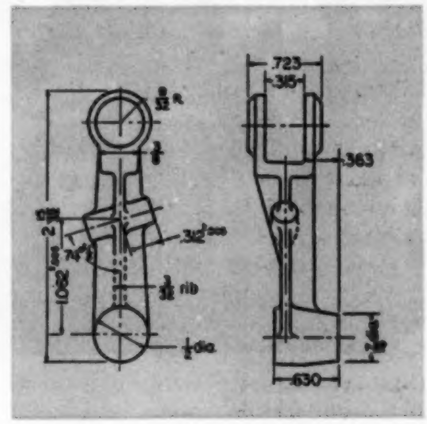


Fig. 16—Right—Aerial camera crank investment-cast in Type 410 stainless steel

and also must be removed. Gating areas necessary for a particular casting cannot always be predicted and, hence, the best practice in design is to indicate those surface areas where gates would be undesirable.

MATERIALS: Practically all metals which can be melted are being cast by the investment process. It not only provides economy for unusual designs in the more common metals but makes possible many otherwise impossible machine parts in metals not machinable to any great degree. All standard cast materials are readily available as are many special alloys such as those of cobalt, nickel and chromium, and vacuum refined copper or pure iron.

Excellent Physical Properties

Physical properties of investment-cast metals are generally equal to the mean between the transverse and longitudinal values for rolled bars of the same metal or alloy. On an average they are about 10 per cent greater than the properties of sand-cast metal. Heat treatable metals respond well without warpage. Any of the tool steels may be used and are often found advantageous in cases where hardened parts must be used. Nitriding steels have been cast but their as-cast hardness is about 400 Brinell and little machining can be done without annealing.

Heat treated aluminum-bronze provides an approximate tensile strength of 85,000 psi with a Rockwell hardness of B-90. Monel "S" or "H" provides a tensile of 120,000 psi and a hardness of Rockwell C-32. Heat treated stainless Type 420 offers the same tensile strength with slightly different properties. Beryllium-copper on the other hand offers tensile strengths up to 170,000 psi in heat treated form. In the steels, heat treated AISI 3140 gives a tensile strength up to 220,500 psi.

Owing to the close control of melt possible, some special castings of controlled composition have been produced by this method. Alloy gray iron pistons, cylinder sleeves, compressor valve sleeves, and other parts have been produced.

TOLERANCES: The investment casting process has often been termed "precision casting", a name which has been misleading at times. Actually, considering large dimensions of more than 5 inches in length, the as-cast tolerances are not substantially better than those found in careful permanent-mold casting or even sand casting. Actual dimensional tolerances on the *average* are specified as plus or minus 0.005-inch per inch for dimensions over 1 inch and plus or minus 0.003-inch on those under 1 inch. The effect of size on tolerances is easily seen.

Some degree of difference is present in casting ferrous and nonferrous alloys when considering tolerances. *Minimum* general tolerances are usually set at plus or minus 0.002-inch per inch for nonferrous alloys and plus or minus 0.004-inch per inch for ferrous. Occasionally small dimensions of ¼-inch or less can be held to plus or minus 0.0005-inch and many parts have been produced with cores, thin slots and shapes held to tolerances of plus or minus 0.001-inch. The problem is not holding the desired accuracy on necessary portions of a part but in many cases is that of holding too high an accuracy on unnecessary portions. Where a gate is removed by grinding, the tolerance ordinarily can be no closer than plus or minus 0.010-inch over the gate area. For a critical dimension across the parting line, it is usually desirable to allow 0.001-inch total in addition to the nominal tolerance per inch allowed.

Some allowance must be made for possible warpage

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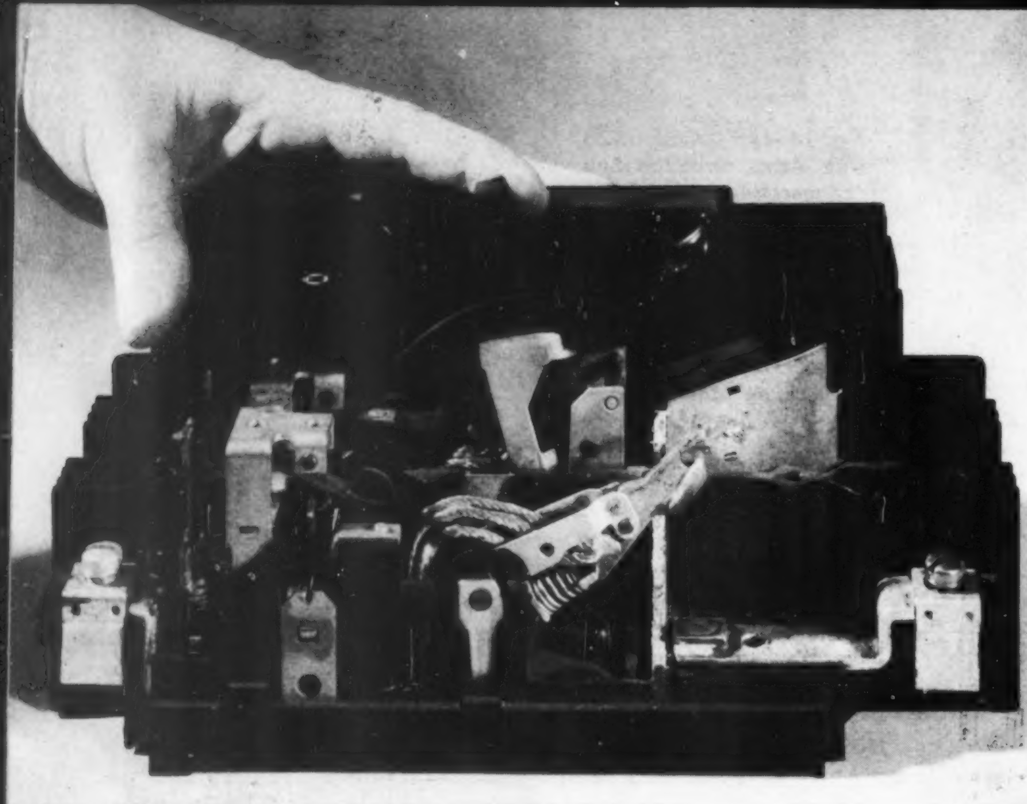

Fig. 1—Cutaway view of breaker showing compactly designed toggle mechanism in open position

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Beaver, Pa.

Overcenter Toggle Improves Design

THREE outstanding improvements in a mechanism for circuit breakers have resulted from the application of the overcenter principle in its design: The physical size is small with respect to current rating, contact pressure actually increases as the contacts wear and the speed of opening is increased, resulting in a higher interrupting rating of the breaker on short circuit.

The small physical size of the breaker, *Fig. 1*, is effected in two ways. The overcenter action produces an additional parting of contacts when the breaker is tripped, a separation that could be obtained otherwise only by increasing the length of the contact arm at least one inch, *Fig. 2*. The construction also allows the contact arm to operate edgewise, decreasing the overall width of the mechanism and hence of the breaker itself.

As the contacts wear, increased contact pressure results from the fact that the toggle mechanism goes farther overcenter. This means that there is no overheating of contacts due to decreased pressure, resulting in a longer life for the unit.

Because of the whip-lash action of the toggle in going overcenter and the shorter contact arm, the speed of contact parting is increased. This rapid action increases the interrupting ability of the breaker on short circuit and the additional separation produced is an important factor in establishing the interrupting rating.

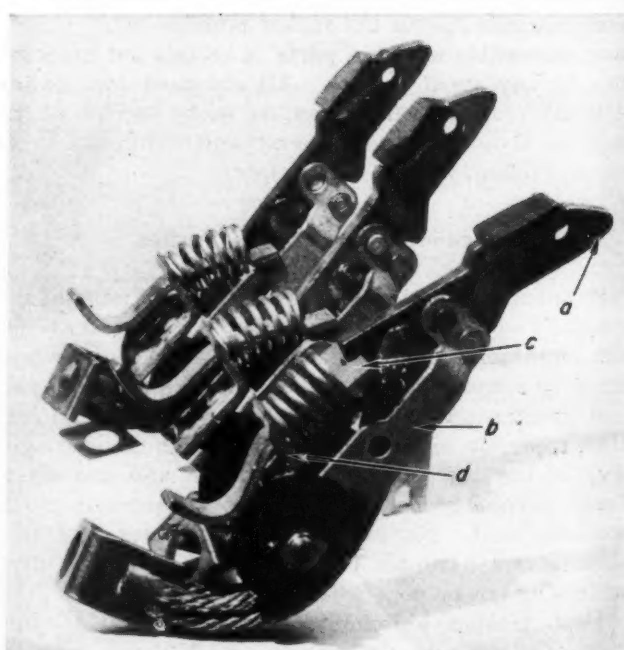


Fig. 2—Overcenter toggle mechanism for a three-pole breaker. Assembly in the foreground is in the tripped position. Current carrying contact arm (a) is pivoted to a metal support (b) which in turn is carried on an insulated yoke. The contact arm pivots on a knife edge on the spring guide (c) which in turn pivots on an L-support (d) and works against a compression spring

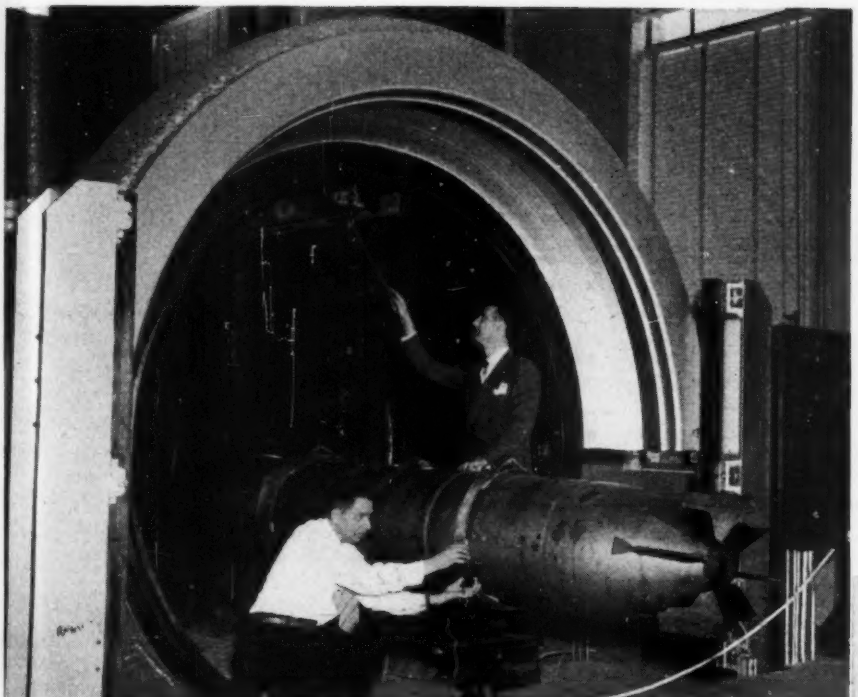
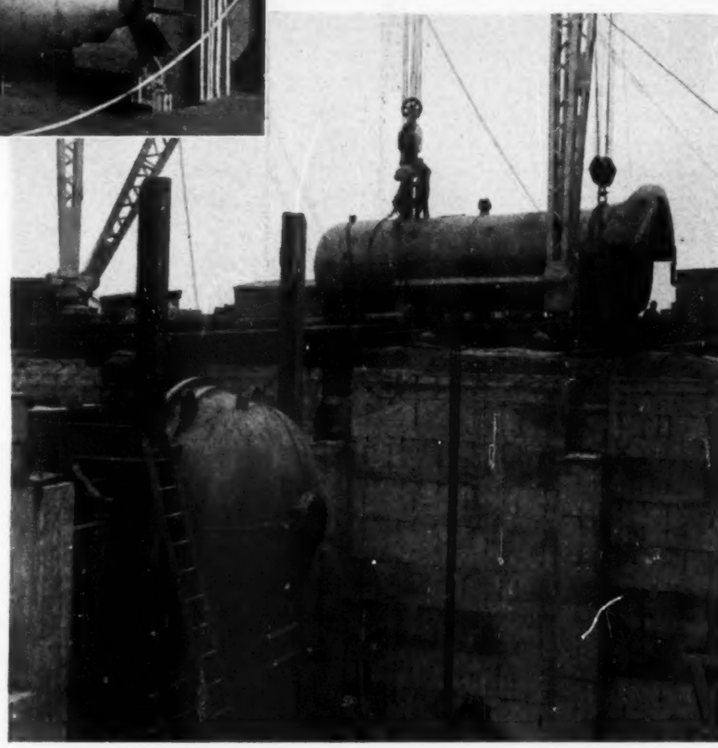


Fig. 1—Left—Front view of 1000-psi testing vessel, showing gasket grooves in shell ring, vertical door guides and control board

Fig. 2—Below—Installation of the huge pressure vessel necessitated construction of specially designed concrete foundation at site. Here the 39-ton door is shown in place, while vessel is readied for hoisting into position



Sealing a 39-Ton Door

. . . against 1000 psi presented extraordinary design problems

By George Mackas
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Naval Ordnance Laboratory
White Oak, Silver Spring, Md.

UNUSUAL problems of design were successfully solved during recent development of the Naval Ordnance Laboratory's huge pressure vessel for testing naval ordnance. The vessel and its quick-opening door, *Fig.* 1, were required to withstand a working pressure of 1000 psi to simulate ocean water pressures down to depths of one-half mile below the surface.

Some idea of the magnitude of this project may be gleaned from the photograph, *Fig.* 2. Having an average inside length of 36 ft 7 in. and a diameter of 8 ft 4 in., the vessel is probably the largest 1000-psi type in the world. Its total internal volume is 1994 cu ft, equivalent to 14,900 gal. The shell alone weighs 111 tons, the door 39 tons, and 61 tons of water are needed to fill the vessel.

Shown in major detail in *Fig.* 3, the vessel consists of a heavy horizontal cylindrical shell, one end sealed permanently and the other closed by a door lifted into position hydraulically. Beneath is a transfer tank in

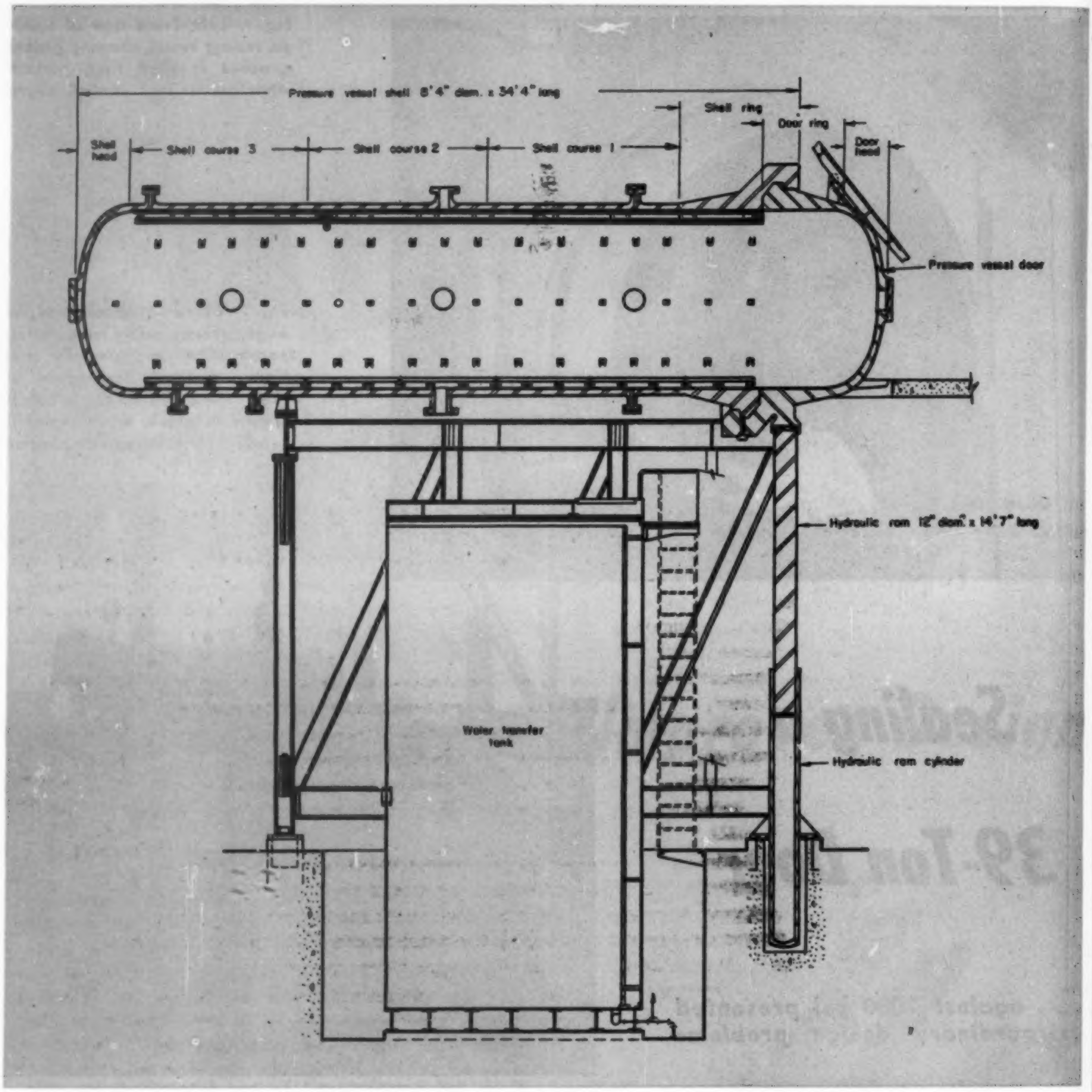

Fig. 3—Section through vessel, door, supporting structure, door-actuating ram, and test-water tank

which water is stored when not needed in the vessel. After the vessel has been filled with water to conduct a test, additional water is pumped in to raise the pressure or is released to lower it. Upon conclusion of a test, the water is drained into the storage tank.

As developed by the laboratory in conjunction with the Babcock & Wilcox and Adamson United Companies, the vessel has a factor of safety of five. The three courses of the shell, which may be seen in the drawing, were rolled from 4 in. thick steel plate, the edges of the courses being pressed hot to obtain the desired curvature prior to welding longitudinally. Girth welding was employed for joining the courses, and also for the hot-spun head at the closed end of the shell and cast-steel ring at the door end. All welding grooves were of the type shown in *Fig.* 4.

Two major difficulties arose in connection with the design of the door, the first being its operation and the second the sealing against leakage. Large doors previously had been made for sealing steam pressure of about 150 psi. These were of the vertical sliding type, using a drum and cable for lowering into closed position. No door of the required size, however, had been built before to withstand 1000 psi. It was evident that the door would be extremely large and heavy, and would require excessive structural facilities if designed for lowering to the shell. The decision therefore was made to reverse the normal method of operation and to lift the door into closed position by hydraulic means.

To support the door and raise it between the two vertical guides, which may be seen in *Fig.* 1, a 12-in.

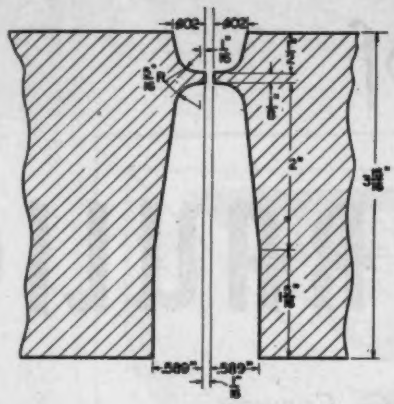

Fig. 4 — Left — Detail of weld groove for all seams of vessel

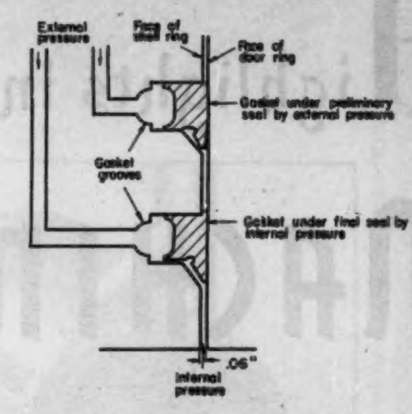

Fig. 5—Right—Schematic illustrates how special rubber gaskets seal door against leakage

diameter forged-steel ram, 14 ft 7 in. long was required, housed in a cylinder 3⅜-in. thick. In moving into position, the face of the door clears the face of the shell ring by not more than 0.06-in., finally coming to rest in the closed position illustrated in *Fig.* 3. It will be noted that the upper half of the door ring fits under and within the upper half of the shell ring, while the lower half fits around it. This design holds the door in place against the total force of 4000 tons resulting from the maximum internal pressure.

Door Raised and Locked Hydraulically

Oil pressure of 840 psi is required to lift the door and is supplied by a Vickers 1000-psi pump driven by a 15-hp motor. When the door is raised to its closed position the oil pump continues to build up pressure until at 900 psi a valve opens and admits oil to a cylinder and pin on each side of the vessel, forcing the pins to move in and lock the door in place. The pins (locking holes for one of which are shown at the left edge of *Fig.* 1) strike a limit switch at the end of their travel, shutting down the pump. As a safety measure, in case the pump should for some reason continue to operate and build up higher pressures, a relief valve opens at 950 psi and diverts the oil back to the drain.

To open the door the oil pump is started again. At 850 psi the oil pressure under the ram lifts the weight of the door off the pins, and at 900 a valve opens to admit oil to the pin cylinders, moving the pins out. At the end of their travel the pins strike limit switches which open another valve, allowing the oil from under the ram to return to the drain. The door then starts down until it reaches its maximum open position where it strikes another limit switch that stops the pump. Operation is, of course, automatic, the door being raised, lowered or stopped in any position from the pushbutton station. Approximately four minutes are required in raising or lowering the door.

In connection with the problem of sealing the door against leakage, previous experience indicated that rubber gaskets would operate without difficulty on vessels subject to 400 psi working pressure and up to 800 psi test pressure. To determine whether they were suitable for 1000 psi working pressure and 2000 psi test pressure, a test plate was made using a gasket 26 in. diameter under conditions similar to those anticipated in the vessel. A maximum pressure of 3000 psi was reached satisfactorily, amply proving that the projected type of gasket would be suitable.

Shown diagrammatically in *Fig.* 5, the gasket is approximately a ¾-in. square with a lip projecting from one side. Two such gaskets are employed, fitted into concentric grooves in the face of the shell ring so that their outer surfaces are flush with the face of the ring. External vacuum is used to pull the gaskets into the grooves while the door is being raised or lowered. As soon as the door closes, positive external pressure is applied to the gaskets, forcing them against the face of the door ring to form a preliminary seal. Internal pressure developed after the vessel is filled with water forces the gasket lip firmly against the face of the door ring, thus effecting a final seal. The external pressure behind the gaskets can be maintained or removed without disturbing the final seal.

Second Gasket Insures Sealing

While it was fully expected that one gasket would be sufficient at all times, a second gasket was provided as a precaution. The gaskets are synthetic rubber, and operating temperature is approximately 70 F. External pressure system for the gaskets consists of a 3-hp Nash vacuum pump having a capacity of 40 cu ft at 15-in. Hg, and an air-operated Aldrich-Lytle power unit with a capacity of 2.2 gpm at 2000 psi with 50 psi air pressure. In order to insure a vacuum behind the gaskets when the door moves up or down, the vacuum pump is interlocked with the oil pump that raises or lowers the door. A further interlock is provided to prevent the door from being opened as long as there is pressure on the gaskets. With the vessel under test at 1000 psi, leakage past the seal was found to be considerably less than the 1 cu ft per hr allowed by the Navy.

Highlights in the History of MACHINE HYDRAULIC

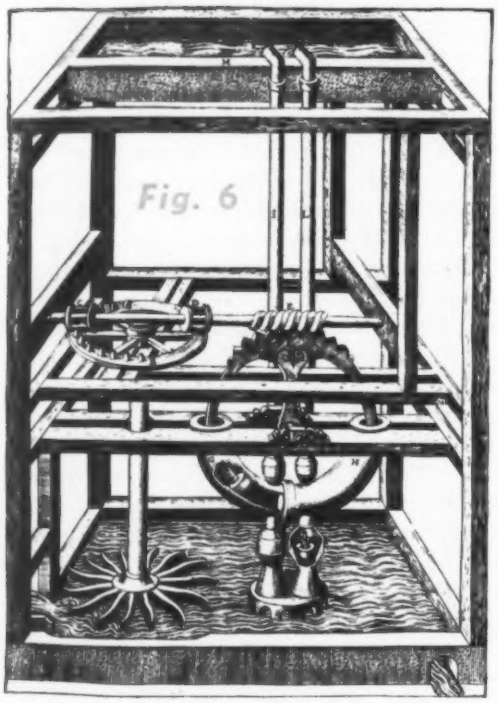

Fig. 6

Illustration, courtesy Science Museum, London

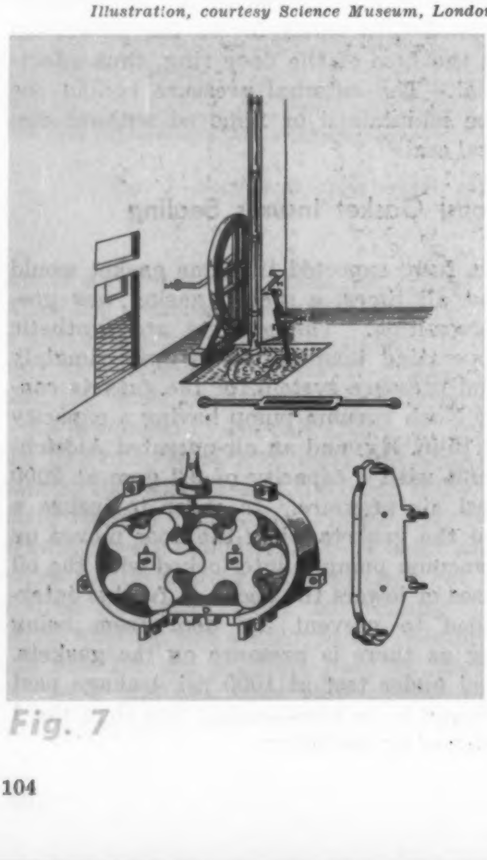

Fig. 7

By H. G. Conway

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Part 3—Rotary Pumps

HISTORY of rotary pumps begins in the sixteenth century and to anyone interested in modern rotary compressors or pumps, the subsequent development in the next 300 years makes fascinating study. The invention of the steam engine turned men's minds to a rotary substitute for the reciprocating engine and an extraordinary variety of mechanisms was evolved and repeatedly reinvented.

What might be described as a semirotary pump is illustrated in *Fig.* 6 from Ramelli's book (1588). This uses a half-contrate wheel to rotate a worm in alternate directions and thus reciprocate pistons in curved cylinders—the construction being obviously impracticable.

The forerunner of the modern gear pump, or of the famous Roots blower of 100 years ago, was a meshing gear pump usually ascribed to Pappenheim. There is some doubt as to whether Pappenheim was a man or a place, but descriptions of the pump indicate that it was known before 1650. Ramelli describes a single gear and lifting-roller vane pump which may have been the forerunner of the Pappenheim pump, since the logical step would be to replace the roller and slide with another gear wheel. *Fig.* 7, redrawn from Serviere (1719), is a fine early illustration of the pump and is remarkable for the ingenious linkage connecting the driving crank to the pump.

Vane pumps, so widely used and known today, are also described by Ramelli although their precise origin is obscure. The design shown in *Fig.* 8, from Ramelli, will be recognized by pump engineers as a modern construction. The partially disassembled pump at the top right shows the swinging vanes. The ordinary loose vane rotary pump, as well as various types of semirotary vane pumps, are described and excellently illustrated.

There must have been many inventors of pumps and engines in the past hundred years who would have benefited from a sight of the illustrations from some of these books. "Three dimensional" pumps, i.e., these based on the sphere instead of the cylinder, are much later developments. Beginning in the nineteenth century, they have not proved to be of practical interest. A good description of such pumps will be found in Reuleaux's *Kinematics of Machinery* (1876).

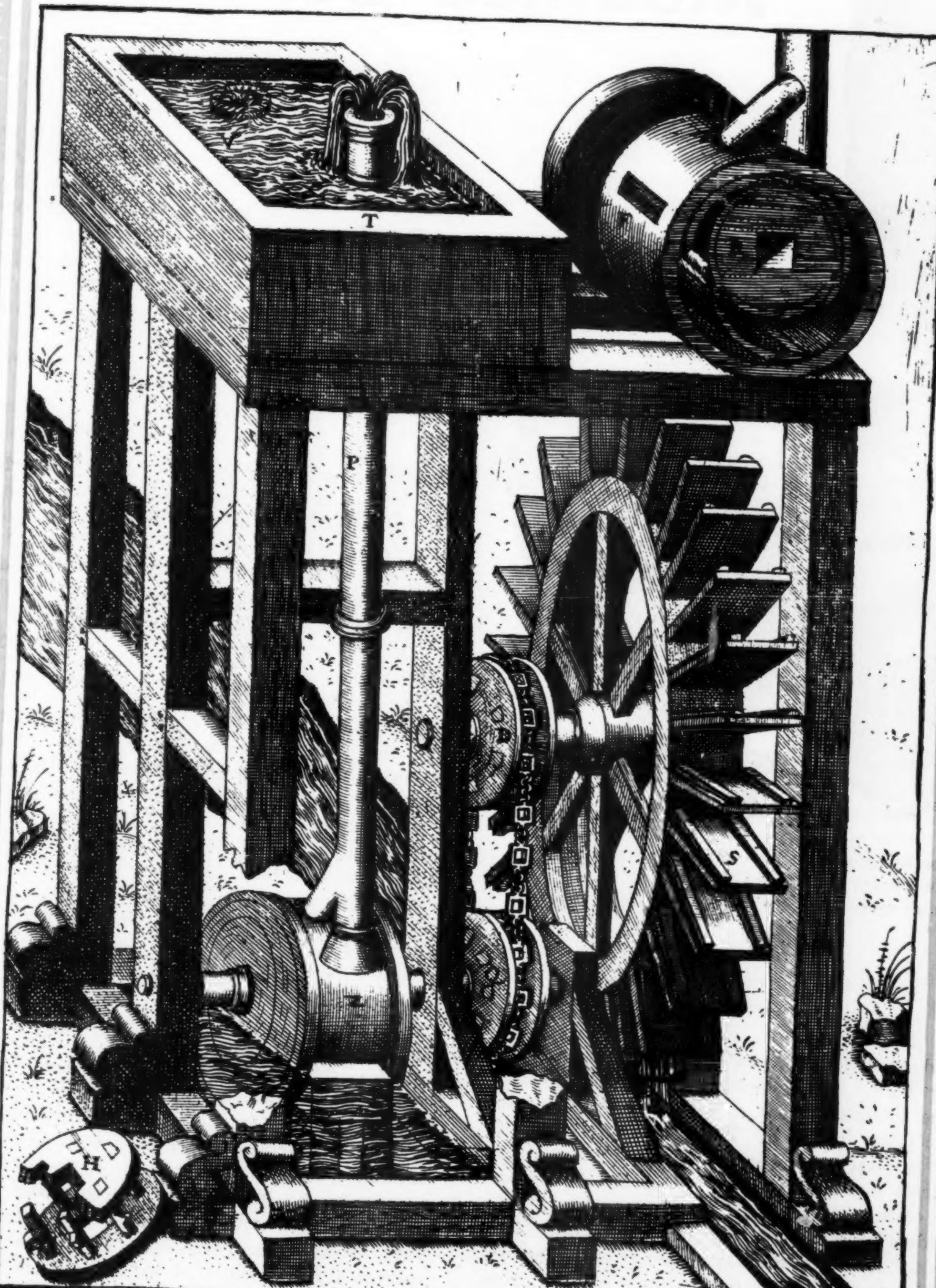

Fig. 8

Illustration, courtesy Science Museum, London

Modified Geneva Driv

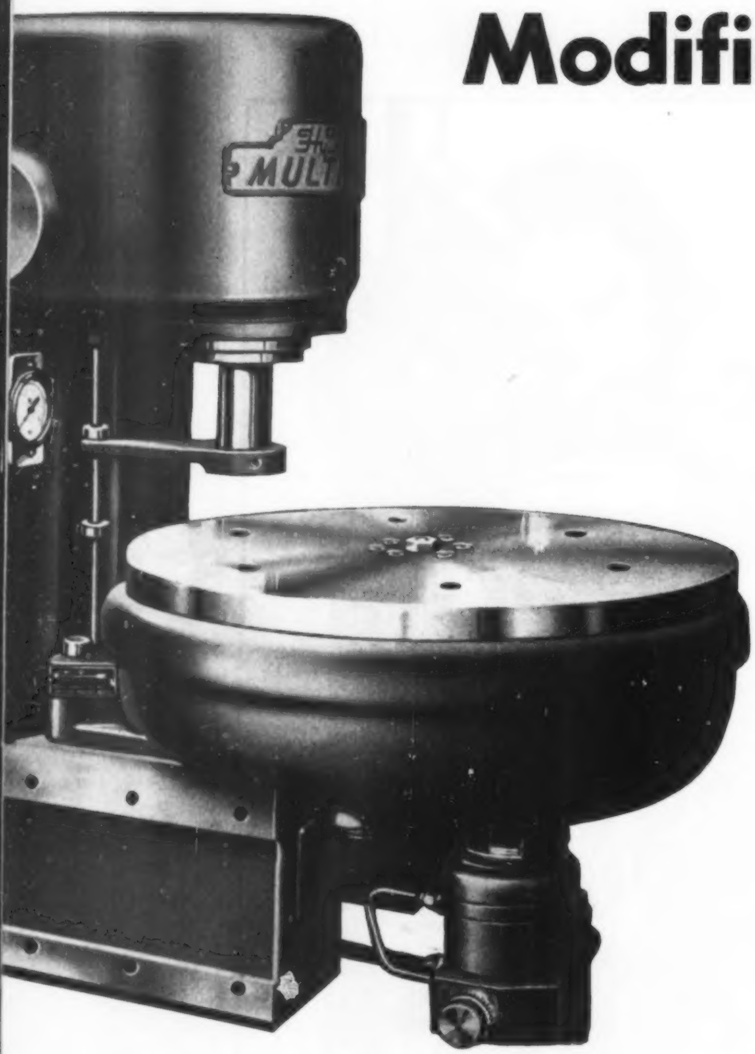

Fig. 1—Operating in sequence with press ram, index table employs modified Geneva drive to effect high cycling rates without backlash

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HIGH cycling rates and elimination of backlash, or jump, during indexing are advantages of the hydraulic index table shown on a Denison hydraulic press in *Fig.* 1. These advances are made possible through the application of hydraulics to the controls and by the use of a Geneva drive mechanism of modified design for the index table.

By hydraulically controlling the index table, it is possible to obtain a greater number of cycles for the machine than with the conventional Geneva drive. Indexing cycles may be varied from 20 to 60 per minute with a $\frac{1}{4}$-inch stroke of the hydraulic-press ram, by adjustment of a valve mounted on the fluid motor under the table.

Unlike the usual Geneva drive in which the driver rotates continuously at a uniform speed, this mechanism of modified design provides for speed variation. By means of the hydraulic control, the driver may be rotated at high speed during the cycle portion that the spider is in locked position. It is also possible to reduce the speed of the driver during its indexing cycle to a rate determined by the part carried on the table or by other limiting factors. The rate of index speed is easily adjusted. However, the rate of speed of the driver in its locking cycle is always at its maximum.

The Geneva drive modification, *Fig.* 2, utilizes projecting spider lugs that extend past the normal outline of a spider of conventional design. The purpose of this projection is to eliminate backlash or jump. Irregular action, characteristic of a conventional Geneva drive, occurs at the time the roller of the driver enters the spider driving slot, and the driver locking face is only half way in the locking pocket of the spider. In the modified spider, however, the extended lugs permit the roller to enter the drive slot at a point just before the actual index. This avoids any possible jerk in the spider motion due to quick pick-up or stop.

The gear mounted on the under side of the Geneva driver has the bottom face machined to an irregular surface. This acts as a cam for controlling a hydraulic valve that governs the speed of the fluid motor as shown in the schematic drawing in *Fig.* 3. The Geneva drive gear meshes with the pinion that is driven by the fluid motor. Flow-control and bypass valve is mounted on the inlet and outlet ports of the motor. The control for the bypass spool in the valve lines up with a tappet that carries a roller actuated by the cam of the Geneva drive gear.

By depressing the valve tappet at the proper time, the gear cam determines the portion of cycle for fast speed and slow speed. The ports are opened fully in the control valve for the high-speed portion and closed to the predetermined volume for the slow-speed. At the end of one complete revolution of the driver,

Speeds Indexing

the valve tappet is further depressed to stop the fluid motor and driver. This automatically interlocks the table with the main valve in the press to start the downward movement of the ram. At any time that the ram is in its downward motion or its upward motion, it is not possible for the table to index. After the ram has completed its upward motion and lifted the press valve shipper rod, the valve automatically starts the fluid motor through its cycle and interlocks the ram in its upward position.

Single or continuous automatic cycling is obtained by the setting of the main valve in the press. For a single cycle, the valve press handle is pulled down momentarily to allow a pressure shot of oil to go to the control valve of the index table and start the cycle. After the table has indexed, the handle may be released and the press will complete its entire cycle, stopping with the ram in the up position. For automatic cycling, the handle is pulled down to the lowest position and then held there by cam action. The press will cycle continuously until the handle is returned to release position.

Operational sequence of a cycle after the valve handle is pulled down is as follows: Oil is directed from the press line through the press valve to the index-table control valve. This starts the fluid motor rotating the Geneva driver for the indexing and locking motions. At the completion of the index and locking cycle, the fluid-motor valve allows a pressure shot of oil to go to the inner spool of the press valve, which when shifted by this pressure causes the ram to descend. The inner spool remains in that position due to line pressure caused by the ram descending. At the end of its stroke, when the ram momentarily stops, and the desired pressure is obtained, the inner spool returns to its original position by spring pressure which overcomes the now reduced line pressure. At the same time, it automatically reverses the flow of oil to return the ram upward. At the top of its stroke, the ram lifts the shipper rod on the press valve and reverts the oil to the fluid motor control valve to start the next index cycle. This will automatically continue until the handle of the press valve is raised to the release position, in which case the press ends its cycle with the ram in its upward position.

It is also possible to employ a two-way skip valve and have it actuated by a cam mounted on the outside edge of the table dial so that, when the dial stops at the end of an index, the cam will depress the skip-valve spool. This action will cause the table to re-index without letting the ram go through its cycle. By spacing a cam on every other station, it is possible to skip every other station and have, in effect, a three-station index table instead of a six. In this manner, the table will index twice for each cycle of the ram.

Fig. 2—Section of Geneva drive mechanism. The modified Geneva spider has extended lug projections that eliminate backlash when the roller enters the spider slot

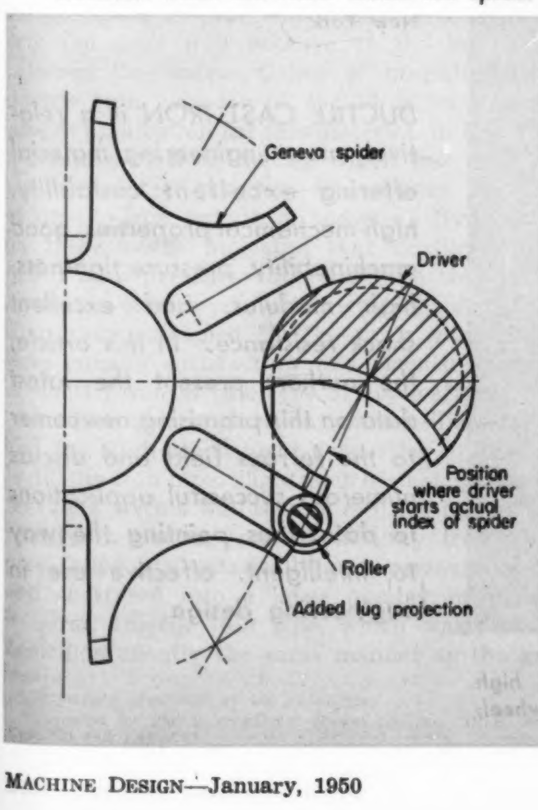
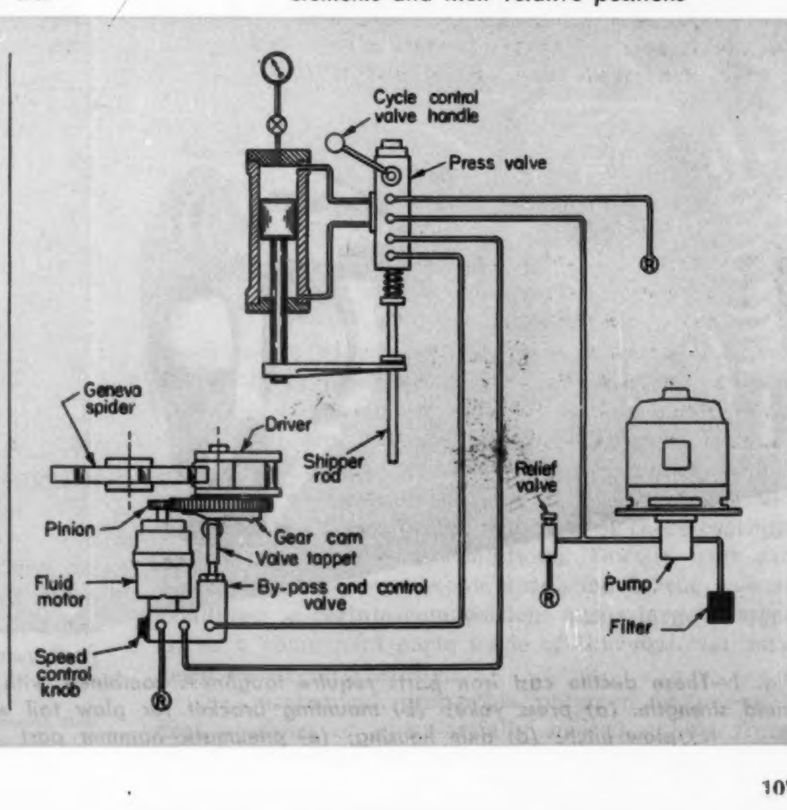


Fig. 3—Schematic drawing of the interlocking circuit for press ram and indexing table, illustrating the operating elements and their relative positions



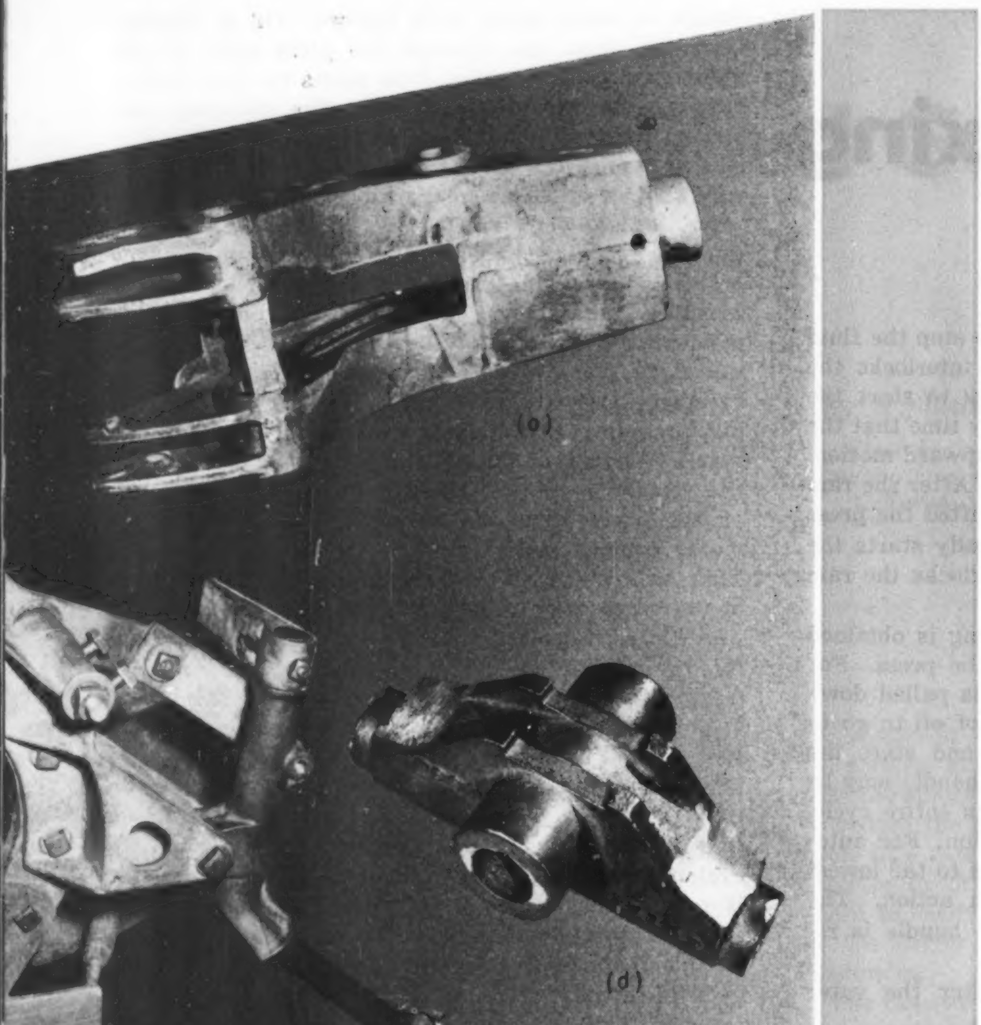


Fig. 1—These ductile cast iron parts require toughness combined with high yield strength: (a) press yoke; (b) mounting bracket for plow tail wheel; (c) plow hitch; (d) axle housing; (e) pneumatic hammer part

engineering applications of

Ductile Cast Iron

By Albert P. Gagnebin, Keith D. Millis and Norman B. Pilling
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DUCTILE CAST IRON is a relatively new engineering material offering excellent castability, high mechanical properties, good machinability, pressure tightness, high modulus, and excellent shock resistance. In this article, the authors present the latest data on this promising newcomer to the ferrous field and discuss numerous successful applications to date, thus pointing the way to intelligent, effective use in engineering design.

SINCE announcement in May, 1948[1] by the International Nickel company of the new process for producing ductile cast iron* by the use of magnesium, interest in this material has mounted steadily in the engineering field. Its superior mechanical properties[2] have resulted in a number of important applications in machinery typical of which are the parts shown in *Fig*. 1.

A high-carbon ferrous material, ductile cast iron contains graphite in the form of spheroids. By contrast, the graphite in ordinary gray iron is in the form of flakes, the difference being strikingly illustrated in the micrographs of *Fig*. 2. The notches and discontinuities of the matrix caused by flake graphite are largely responsible for the relatively low strength and toughness of ordinary gray iron. Internal notching is at a minimum in ductile iron which has a greatly increased proportoin of effective matrix. Thus, the latent qualities of the matrix can be more fully realized with graphite in spheroidal form and a wide variety of superior mechanical properties, including high ductility and toughness, can be achieved.

CASTABILITY AND PRESSURE TIGHTNESS: Ductile iron enjoys the same process advantages that have made gray iron a popular and widely used casting material for so many centuries. It has a low melting point, exceptional fluidity and good machinability. It is particularly suitable for producing complex, intricate shapes requiring mechanical properties of a high order. Many parts are too intricate to be cast in steel and have inadequate mechanical properties when cast in gray iron.

Pressure castings in general, regardless of their intricacy, offer attractive possibilities for ductile iron because, if properly fed, there is no doubt of their pressure tightness. The spheroidal shape of the graphite precludes leakage which may occur even in properly fed gray iron because of the interconnections between the coarse flakes of graphite. The heavy ductile iron cylinder of *Fig*. 3 is an example of a proper application for this material. In *Fig*. 4 is shown a compressor head casting which was used in an experimental test program to compare the properties of ductile iron, gray iron and steel. Hydrostatic tests on these heads indicated that ductile iron casting failed at a pressure comparable to that required to burst the steel head, which was considerably greater than that sustained by the gray iron. Other tests were equally satisfactory and promise extensive use of ductile iron in this type of application.

Another new application which utilizes the fluidity and the high mechanical properties of this material is ductile iron pipe. Eight-inch diameter pipe, eighteen feet long, with a wall section of 0.420-inch, was cast as readily in ductile iron as in gray iron. Gray iron pipe failed hydrostatically at a pressure of 2400 psi and shattered into a large number of pieces when it burst. Ductile iron pipe, which was cast and annealed in exactly the same manner as the gray iron pipe, expanded under pressure and developed a pronounced bulge in one zone. Failure occurred at a pressure of 5300 psi in the form of a longitudinal crack along the bulge and with no shattering. It is obvious that higher pressures can be used in ductile iron pipe and that less maintenance will be required because of fewer failures due to ground shifts and water hammering. Furthermore, it is anticipated that its ductility will permit its use for the transmission of certain fluids and gases which are now barred to cast iron because of its lack of ductility.

TOUGHNESS: A large and obvious area of application for ductile iron is in castings requiring moderate impact or shock resistance. In the past, many cast iron parts have failed because of shock loads in normal service. Numerous component parts on agricultural, road maintenance, and earth moving equipment are in this category. Examples are tractor front-wheel forks, rear-wheel hubs, transmission cases, axle housings, mower guards and points, starter housings, levers, pulleys and gears.

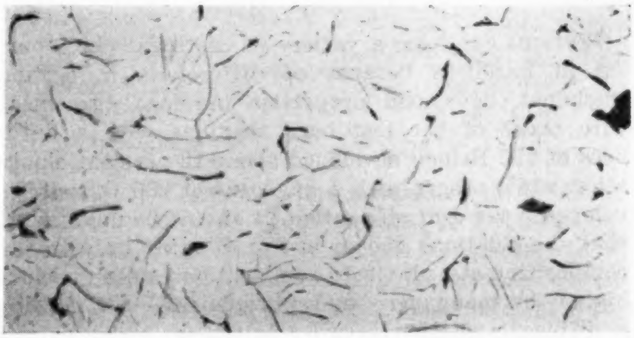
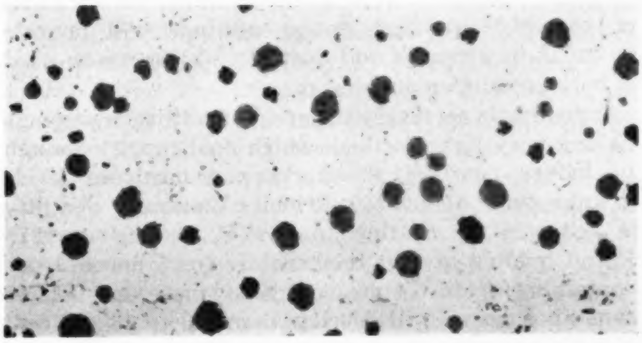


Fig. 2—Graphite structures of gray iron (above) and ductile cast iron (below)



It is the opinion of some engineers that a toughness level represented by 5 per cent elongation would prevent the impact failure of many of these castings under normal service conditions. Ductile iron can readily achieve 5 per cent elongation in the as-cast condition in certain compositions and a large number of such component parts made of this material have been tested successfully.

The mechanical properties obtained in one-inch test sections of commercially melted cupola irons

[1] References are listed at end of article.

* Patented by The International Nickel Co. Inc., U. S. Patents Nos. 2,485,760 and 2,485,761.

from a composition range favorable for as-cast ductility are shown in *Fig.* 5. These data show that a relation exists between the hardness, strength and ductility of ductile iron. Hardness in the test blocks was varied by stripping them from the molds at various temperatures and thus achieving different cooling rates through the critical range. The ductility of this material is a function of the proportion of ferrite and pearlite in the structure. Slow cooling promotes ferrite structures with high ductility while rapid cooling promotes pearlitic structures of higher hardness and low ductility.

The data in *Fig.* 5 show that ductility of 22 per cent elongation has been achieved in the as cast condition. This remarkable value is higher than normal and is the result of a number of favorable circumstances. However, many foundries regularly produce ductile iron having 8 to 15 per cent elongation in one-inch sections in the as-cast condition. Such irons have a microstructure containing a large proportion of ferrite.

Cast and Test Properties Comparable

Sections cut from a variety of castings, which varied in hardness because of differences in section thickness, developed properties in close agreement with those of the test bars. Sections with a hardness of 195 Brinell developed about 12 per cent elongation while others with a hardness of 230 Brinell developed 5 per cent elongation. It should be understood that the relations shown in *Fig.* 5 apply only to the composition stated. Other compositions, notably those higher in manganese and phosphorus, will develop a different relation between hardness, strength and ductility. When sufficient test results have been accumulated to establish the hardness-ductility-strength relationships of various compositions, a simple method of quality control will be available since the hardness of properly produced, sound castings will provide an index of strength and ductility which can be used in both production and design.

Annealing is necessary to develop ductility in castings having very light sections which cool rapidly enough to induce pearlitic structures and hardness levels greater than about 280 Brinell. Maximum ductility is achieved by heating to 1600 F, cooling to 1275 F and holding at that temperature for 5 hours. Irons containing up to 0.4 per cent manganese and 0.1 per cent phosphorus will develop about 18 to 23 per cent elongation after this treatment, while those containing up to 0.8 per cent manganese and 0.20 per cent phosphorus will develop about 15 per cent elongation. Considerably broader composition limits can be tolerated if heat treatment is to be used to achieve ductility than if it is to be obtained in the as-cast condition. If maximum ductility is not required, a simple treatment at 1300 F will readily develop about 10 per cent elongation.

Fig. 3—High mechanical properties and assured pressure tightness make ductile cast iron ideal material for this hydraulic press cylinder

The parts shown in *Fig.* 1 are examples of castings that require a certain degree of toughness in combination with high yield strength for their successful performance. The casting in *Fig.* 1*a* is a component part of a press and is used in the as-cast condition. Typical farm implement parts, also used in the as-cast condition, are the tail-wheel mounting

Table 1—Compression Properties of Ductile Cast Iron

Mark	Composition (%) C	Si	Ni	Mn	P	Condition	Yield Strength* (psi)	Compressive Strength** (psi)	Brinell Hardness Number	Tensile Strength (psi)
A	3.1	2.2	2.8	0.8	0.06	As-cast	94,000	198,000	347	77,600
B	3.1	2.3	1.9	0.8	0.06	As-cast	88,400	193,400	306	92,300
C	3.6	2.3	1.9	0.4	0.03	As-cast	76,600	174,200	277	108,700
D	3.5	2.3	0.9	0.2	0.03	As-cast	57,400	125,600	219	80,500
E	3.6	2.3	3.5	0.2	0.04	As-cast	95,600	178,800	321	123,000
F	3.6	2.3	3.5	0.2	...	Annealed	66,000	122,800	195	72,300

* Deflection, 0.5 per cent under load.
** Not true compression values. Specimens bowed after the stress exceeded the yield point and final failure occurred by shearing.

Fig. 4—Hydrostatic tests on ductile cast iron compressor heads such as this indicated strength on par with that of a steel head of similar design

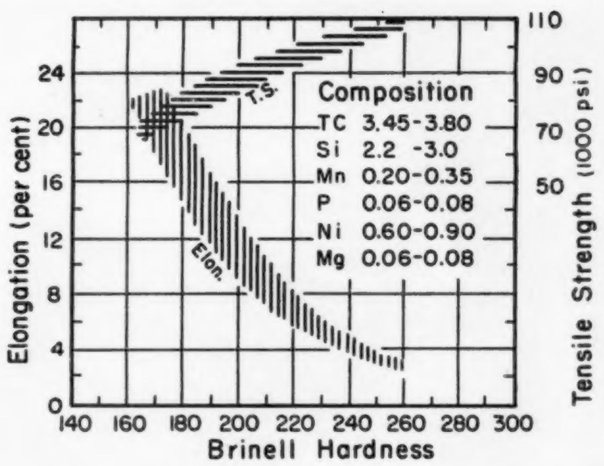


Fig. 5—Relationship between elongation, tensile strength and hardness of ductile cast iron in one-inch sections as-cast

bracket and the plow hitch shown in *Figs.* 1*b* and *c* respectively. More than a thousand of the axle housings shown in *Fig.* 1*d* have been produced and are giving satisfactory service on road equipment machinery. *Fig.* 1*e* shows a component part for a pneumatic hammer which is used in the annealed condition because it is subjected to heavy shock loads.

Although considerable emphasis has been given ductile irons with predominantly ferritic structures, the possibilities of other structures should not be overlooked. The new material can be produced with a variety of structures such as pearlitic, acicular, martensitic, and austenitic, each of which has its own area of application. Examples of parts produced with pearlitic structures to obtain wear resistance are the bevel gears of *Fig.* 6*a* and the worm and worm wheel of *Fig.* 6*b*. Frequent failures occurred when the bevel gears were made of gray iron, whereas no failures have been experienced with the 300 sets of ductile iron gears tested in field service. A large number of worm and worm wheel assemblies have been produced from ductile cast iron having the following typical properties:

Yield Point	75,000 psi
Tensile Strength	103,000 psi
Elongation	3.7 per cent
Brinell Hardness	250

Rigidity: Ductile iron has a modulus of elasticity of about 25,000,000 psi. This value is virtually unaffected by composition or section thickness and is uniform from heat to heat. The material is elastic in the generally accepted sense and develops proportionality of stress to strain up to high loads.

It is well known that gray cast iron does not follow Hooke's law and its apparent modulus steadily decreases with increasing stress. The average stiffness varies with the amount of graphite present and ranges from 12 x 10^6 psi in high-carbon iron to 20 x 10^6 psi in low-carbon iron. The modulus of gray cast iron is also a function of the shape and distribution of the graphite flakes so that some variation may be encountered in heats of similar composition.

Stiffness is an important feature in certain engineering applications of which paper mill rolls are an example. Such rolls are machined with a cambered surface to overcome the deflection resulting from their weight and from working loads in order to produce a uniform gage across the width of the paper. A uniform modulus of high value in combination with high strength is advantageous in these circumstances.

Aside from its high mechanical properties, which

Table 2—Gall Test Data Comparing Ductile and Gray Cast Iron

Type of Iron	Type of Graphite	Composition (%) C	Si	Ni	Mn	P	S	Tensile Strength (psi)	Brinell Hardness Number	Scoring Load (lb)	Galling Load (lb)	Maximum Apparent Stress (psi)
Gray	Fine Flake Dendritic	3.1	1.7	2.0	0.8	0.09	0.09	30,000	224	150	190	41,000
Gray Inoculated	Coarse Flake	3.0	1.8	2.7	0.7	0.09	0.09	36,000	218	300	400	59,000
Gray High Phos.	Coarse Flake	3.1	1.7	2.1	0.7	0.46	0.13	47,000	241	375	560	66,000
Ductile	Spheroidal	3.0	1.7	1.9	0.9	0.02	0.01	90,000	279	400	500	71,000
Ductile	Spheroidal	3.0	2.2	2.0	0.2	0.02	0.01	80,000	217	300	400	54,000
Ductile	Spheroidal	3.5	2.3	2.0	0.2	0.02	0.01	.....	212	400	500	59,000

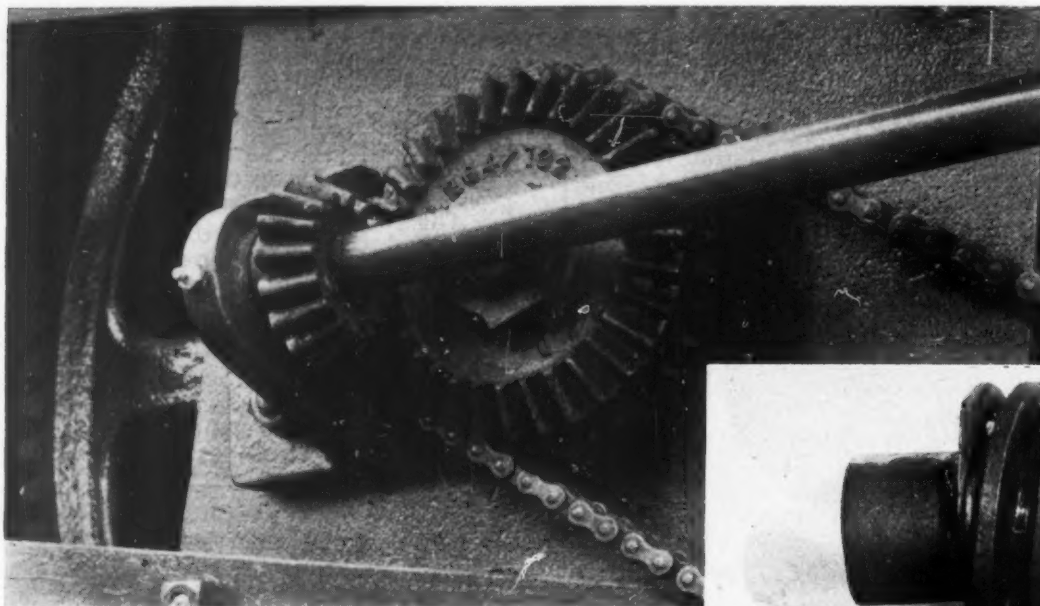


Fig. 6—To obtain high wear resistance, these parts were produced of ductile cast iron having pearlitic structure: (left) bevel gears used on hay baler; (below) worm and wheel used in rail car equipment

will be useful in many other types of rolls, a less obvious advantage of ductile iron is the superior quality of its machined surface. Some rolls require a flawless surface which is difficult to achieve in other cast ferrous materials.

ABRASION RESISTANCE COMBINED WITH TOUGHNESS: A unique property of ductile iron is that it can combine a hard, chilled surface with a ductile core or back. By appropriate composition control and casting the material against a metal chill, it is possible to combine a carbidic, abrasion-resistant surface with a moderately ductile back. Tempering treatments below the critical range not only toughen the chilled surface but also improve the ductility of the material adjacent to the chilled zone. Combinations of 52 to 60 Rockwell C hardness on the chilled surface, with 2 to 10 per cent elongation in the supporting metal, can be achieved by this method. No other material can achieve this combination of structure and properties. Its only counterpart is a bimetal consisting of a tough alloy coated with a hard surfacing material.

One remarkable application of this unique ability is in plow shares, *Fig.* 7. About 25,000 of these shares have been produced with an unconditional guarantee on breakage. The ductile iron shares have shown impressively better abrasion resistance than other commonly used materials.

COMPRESSION PROPERTIES: The compression properties of a few ductile cast irons, recorded in TABLE 1, indicate that the compression strength varies directly with hardness. The yield strength ranges from 57,000 psi in iron having a hardness of 219 Brinell to 95,000 psi in iron with a hardness of 321 Brinell. The materials develop proportionality of stress to strain up to the proportional limit.

The ultimate compression strength values given in TABLE 1 are believed to be low because the specimens bowed after the yield point was exceeded and eventually failed in shear. These specimens were 4.2 inches long and had a 0.798-inch diameter reduced section for a length of 2.4 inches. Specimens 1 inch in diameter by 1 inch long were machined from the soft irons designated as *D* and *F* in TABLE 1. They were loaded to the limit of the capacity of the testing machine, which was equivalent to a stress of 225,000 psi, and were reduced from the original length of one inch to 0.602 and 0.550 inch respectively.

GALL RESISTANCE: The excellent wearing quality of gray cast iron is well known and is generally attributed to the presence of graphite in its structure. Considerable interest attaches, therefore, to the effect of changing its form from flake to spheroid and its probable influence on resistance to rubbing friction.

A gall test, developed by the research laboratory of the International Nickel company consists of rubbing a cylinder against an annular ring of the same material without lubrication. The cylinder is oscillated 15 degrees 11 times a minute, and the load between the specimens increased by 10 pounds during each oscillation. Tests are made to progressively higher loads, bracketing the loads that cause scoring and those that cause galling. The apparent stress developed is calculated by estimating the area of

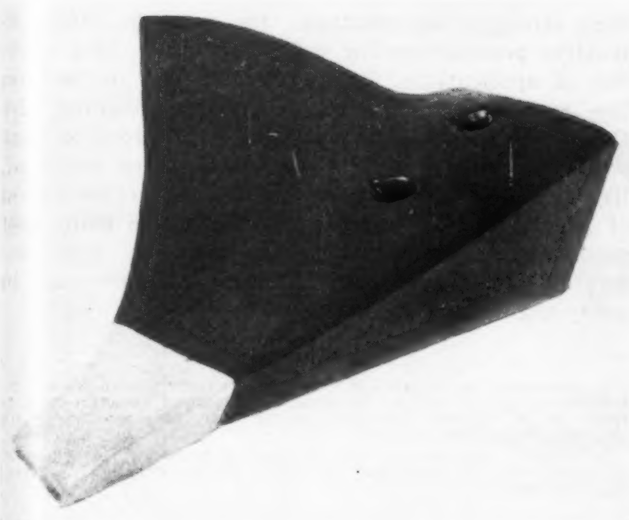


Fig. 7—Combination of hard, chilled surface with ductile core gives ductile cast iron plow shares superior abrasion and breakage resistance

Fig. 8—Ductile cast iron hammer anvil weighs 40,000 pounds and has section thickness of 36 inches

contact on the cylinder and dividing into the load.

Data in TABLE 2 demonstrate the well-known fact that gray irons with coarse flake graphite are considerably superior in gall resistance to iron containing fine, dendritic graphite. Galling occurred at 190 pounds load with dendritic graphite iron and at 400 pounds load in iron of similar composition but containing flake graphite.

The ductile irons with spheroidal graphite, particularly the one with a pearlitic matrix and hardness of 279 Brinell, do not appear to be inferior to the coarse flake graphite iron. The soft ductile irons, of 220 Brinell hardness and with a ferritic-pearlitic matrix, were substantially superior to the dendritic graphite gray iron. These results indicate that the wearing quality of ductile cast iron should be equivalent to that of gray cast iron. However, no operating data are available as yet to support this conclusion.

MACHINABILITY: Turning tests which were made with Carboloy tools to compare the machinability of gray iron and ductile iron are summarized in TABLE 3. These results show that ductile iron with a hardness of 220 Brinell and containing 3.6 per cent carbon and 2.9 per cent silicon had the same tool life as gray iron containing 3.2 per cent carbon and 2.29 per cent silicon and having a hardness of 143 Brinell. When the ductile iron was annealed to 170 Brinell it had a machinability rating of 180 as compared to 100 for the gray iron. A spheroidal graphite iron containing 3.2 per cent carbon and 2.3 per cent silicon, which was included for comparison even though it does not represent the optimum composition for ductile iron, had a lower machinability rating than the gray iron. The ductile irons consumed more power during machining than the gray iron even when they had the same or better machinability ratings, which is perhaps an indication of greater toughness. The ductile irons developed longer chips during machining than the gray iron and use of cutting fluid had no appreciable effect on machinability.

Thus, these tests indicate that ductile cast iron of optimum composition has the same machinability as low-strength gray iron in the as-cast condition and is far superior in the annealed condition. Another feature that has been widely observed is that the

Table 3—Machinability Results of Flake vs. Ductile Cast Iron[1]

					Tool Life				
Material	Composition (%)	Condition	Brinell Hardness Number	Cutting Fluid	V10 (cutting speed, fpm, for 10-minute life)	% V10	T600 (tool life, minutes, at 600 fpm)	Power (net hp/cu in./minute	% Power
Gray Iron	C 3.2, Si 2.2, Mn 0.6, S 0.04, P 0.15	As-cast	143	Dry Emulsion	440 555	100 126	4.4 8.0	.55 .55	100 100
Ductile Iron	C 3.2, Si 2.3, Mn 0.6, Ni 0.50, P 0.15, Mg 0.06[2]	As-cast	248	Dry Emulsion	305 325	69 74	1.5 1.6	.77 .76	140 138
Ductile Iron	C 3.6, Si 2.9, Mn 0.35, Ni 0.65, P 0.05, Mg 0.08	As-cast	210	Dry Emulsion	451 451	102 102	4.3 4.3	.74 .73	134 132
Ductile Iron	C 3.6, Si 2.9, Mn 0.35, Ni 0.65, P 0.05, Mg 0.08	Annealed	170	Dry Emulsion	780 780	177 180	22.7 23.1	.69 .65	126 118

[1]Machine used—14-in. lathe; tools—Carboloy 883 carbide; depth of cut—0.050-in.; feed—0.0129-in. per revolution.
[2]Estimated.

quality of the machined surface of ductile iron is better than that of gray iron and, in fact, resembles that of steel.

An extensive number of ductile iron castings have been produced including hammer anvils weighing 40,000 pounds with a section thickness of 36 inches. One of these is shown in *Fig.* 8. One foundry has melted as much as 100 tons each day for several successive days. Other smaller shops are regularly producing 4 to 5 tons each day. There appear to be no limitations to the ductile iron process with respect to the melting unit used, quantity of metal to be treated, or the intricacy or size of castings.

Performance data are being accumulated in a number of engineering fields and, as with any new engineering material, time is required to evaluate the material and make the most effective use of it. Its high strength and stiffness, for example, offer attractive possibilities for weight reduction in a number of applications. Interest so far has focused on the as-cast and annealed properties. However, the flexibility in properties available with alloys or heat treatment should not be overlooked. This material, like steel, is capable of developing a wide combination of properties. Its superior performance in many test applications, coupled with its moderate cost, indicate that it is destined to find extensive use in many engineering fields.

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Standard Reducers Permit Varying Torque and Speed

IN A POWER bender, the operating specifications most subject to changing requirements are torque and speed. These factors are made easily variable in the bender shown in the accompanying illustration which is made by the Hancock Tool and Die Corp., Detroit, Mich. Basic operating components of the machine are a rotary fixture drive and a pressure cylinder arrangement by which the work is bent

to the shape of the fixture as the fixture revolves. The fixture drive is composed of a motorized, self-contained standard speed reducer, rather than built-in gearing. This makes it possible to alter the speed and torque of the basic machine simply by an interchange of speed reducers and motors.

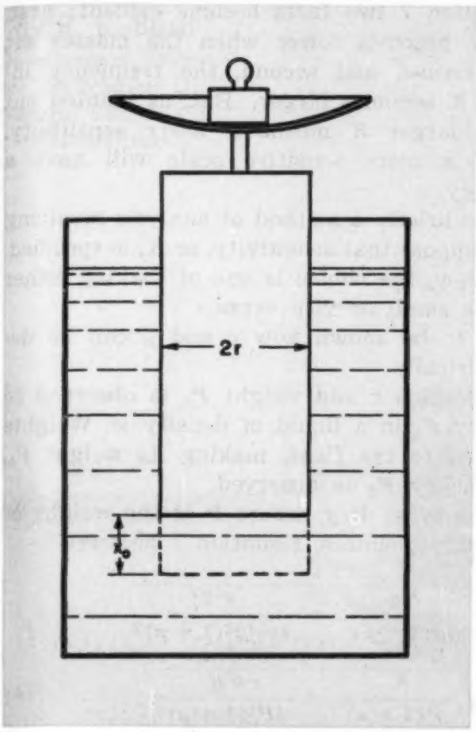
Torque requirements for a power bender of this type are of course quite high. The machine shown, for example, has a torque capacity rating of 44,000 in-lb and employs a 15-hp, 1200-rpm motor. To meet these requirements, plus a reserve capacity, and still keep the machine design compact, standard Cone-Drive geared reducers are used.

Minor variations in speed—between the various ratios of standard Cone-Drive reducers—are obtained by proper selection of the pulley drive between motor and reducer input shaft. The output shaft of the reducer is coupled directly to the rotating fixture table. The latter is so designed that a wide variety of fixtures can be readily interchanged on the machine.

The Cone-Drive reducers are of the double-extended gear shaft type, permitting use of the reducer to time the machine through a cam plate mounted on the lower end of the output shaft and actuating a series of electrical switches and solenoids to control the air valves to the pressure cylinder, the motor brake, etc. Interchange of cams for different fixtures is thus also made quite simple. The motor brake is located on one end of the motor armature shaft to stop the fixture rotation at specified points in the bending operation.

In operation—which is automatic, either forward or reverse—the operator clamps one end of a piece of rolled stock or tubing in the fixture and presses the start button. As the fixture revolves, a shoe is forced against the stock by an air cylinder, forming the strip to the external contour of the fixture. The machine stops automatically permitting the operator to unclamp and remove the finished part.

"Today, mechanisms of greater complexity and precision than even before are mass produced by men who usually do not understand the exact function of the part they manufacture. Yet successful operation of the complex precision machines is wholly dependent upon precise control of surface finish. The task of determining such exact surface finishes and of clearly designating them to machine operators now belongs to the designer"—JAMES A. BROADSTON, *North American Aviation Inc.*



Weighing Speed

. . . of hydraulic scales predicted by simple analysis

By Sigmund Rappaport

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SPEED at which a weighing unit can be operated, and to a degree the overall effectiveness of the machine on which it is used, is governed by the scale's oscillation time. Defined as the period required by the scale to react to the load, oscillation time is related to scale sensitivity. Theoretical consideration of this relationship predicts closely the actual performance of a scale, as will be shown in the following development.

If the equilibrium of a hydraulic scale is disturbed by the application of a load the float, *Fig.* 1, which is assumed to be cylindrical in this discussion, will oscillate about the point of equilibrium. Viscosity, inner friction, turbulence, etc., the combined effect of which is called damping, will cause the amplitudes of the oscillations to become progressively smaller until the movement subsides completely. If the damping effect is critical, no marked oscillation occurs, there being instead a slow creeping back to the point of equilibrium. Only damping *below* the critical point will be considered in this development.

The time-amplitude curve of the oscillation is similar to that illustrated in *Fig.* 2. Equation of motion of this damped free vibration is*

$$x = e^{-ct/2M}\left(C_1 \cos t\sqrt{\frac{K}{M} - \frac{c^2}{4M^2}} + C_2 \sin t\sqrt{\frac{K}{M} - \frac{c^2}{4M^2}}\right) \qquad (1)$$

where

x = Amplitude, inches, at any given time t
e = Basis of natural logarithms = 2.71828
c = Damping constant, ounce-seconds per inch
M = Whole mass set in motion during oscillation, ounces
t = Time, seconds, measured from the beginning of free vibrations
K = Force required to lower the float one inch, ounces per inch
C_1 = Integration constant
C_2 = Integration constant

The amplitude of oscillation is progressively diminished by the application of a force proportional to the float velocity and in the opposite direction. This force can be denoted as c times the velocity. An empirical constant, c, must be determined experimentally in each special case, as will be shown later. Its dimensions are force ÷ velocity = oz sec per in.

The mass M consists of two parts:

1. Constant mass of the float, m, including stem, tray and load, which can be replaced by the mass of the liquid displaced in the equilibrium position
2. Variable part of the liquid set in motion which adheres to the float and depends on such factors as shape of float, ratio of vessel and float diameters, viscosity, velocity, etc.

The second part varies in mass and acceleration during the oscillation. To the present time it has been impossible to evaluate this quantity theoretically in a general way. Considering the relatively small amplitudes of a hydraulic scale, it appears feasible to assume this mass to be of constant average value $m\mu$, assigning to it the same acceleration as the float. The equation for the entire mass is then written

Fig. 1—Top—Essentials of hydraulic scale with maximum amplitude of oscillation, x_0, indicated

*J. P. Den Hartog—*Mechanical Vibrations*, McGraw-Hill Book Co. Inc., New York, 1934, Pages 35-54.

$$M = m(1 + \mu) \quad \text{(2)}$$

where μ must be found experimentally, as shown later. Dimensions of the masses are taken to be ounces divided by g = oz sec^2 per in. Earth acceleration g is equal to 386 in per sec^2.

The force K required to lower the float one inch is equal to the weight of the liquid displaced by the one-inch movement:

$$K = r^2 \pi w \quad \text{(3)}$$

where r = float radius, inches, and w = unit weight of liquid in oz per in.[3] This force is equivalent to the commonly used "sensitivity reciprocal," scale sensitivity and K being inversely proportional. Dimensions of K are oz per in. It should be noted that the roots of Equation 1 can never become zero or imaginary since $c^2/4M^2$ is always less than K/M, the oscillation being a damped one *below* critical.

In order to simplify notation,

$$q = \sqrt{\frac{K}{M} - \frac{c^2}{4M^2}} \quad \text{(4)}$$

Referring to Equation 1, it can be readily shown* that $C_1 = x_0$, the maximum amplitude of oscillation

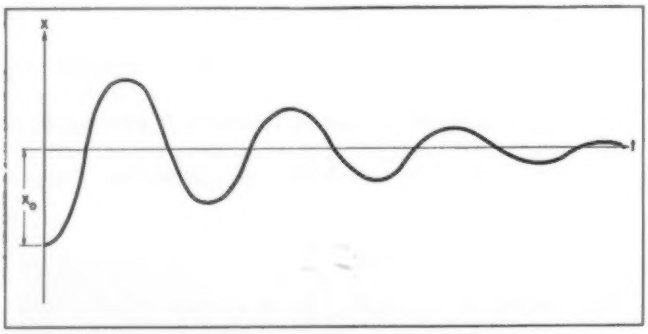

Fig. 2—Time-amplitude curve of damped-free vibration

at $t = 0$, and $C_2 = cx_0/2Mq$. Substituting these values and Equation 4 into Equation 1,

$$x = x_0 e^{-ct/2M}\left(\cos qt + \frac{c}{2Mq}\sin qt\right) \quad \text{(5)}$$

The bracketed value in Equation 5 repeats itself when qt moves through 2π. If the duration of this cycle is noted as T, then $qT = 2\pi$ and $T = 2\pi/q$, which is the time required for a full oscillation comprising one up and down stroke. The equivalent frequency, F, or the number of full oscillations per minute, would then be

$$F = \frac{60q}{2\pi} = \frac{30q}{\pi} \quad \text{(6)}$$

Substituting Equation 2 into Equation 4 then placing the latter into Equation 6

$$F = \frac{30}{\pi}\sqrt{\frac{K}{m(1+\mu)} - \frac{c^2}{4m^2(1+\mu)^2}} \quad \text{(7)}$$

* J. P. Den Hartog—*Mechanical Vibrations*, McGraw-Hill Book Co. Inc., New York, 1934, Pages 35-54.

From Equation 7 two facts become evident: first, the frequency becomes lower when the masses set in motion increase, and second, the frequency increases when K becomes larger. But, as pointed out previously, a larger K means a lower sensitivity. Consequently, a more sensitive scale will have a lower frequency.

To illustrate briefly a method of analysis involving Equation 7, suppose that sensitivity, or K, is specified. Since $K = r^2\pi w$, the choice is one of making either r large and w small or vice versa.

It remains to be shown how c and μ can be determined empirically.

A float of radius r and weight P_1 is observed to have frequency F_1 in a liquid of density w. Weights are then added to the float, making its weight P_2, and the frequency F_2 is observed.

Inasmuch as $m = P/g$, where P is the weight of the float assembly, ounces, Equation 7 becomes

$$F = \frac{30}{\pi}\sqrt{\frac{Kg}{mg(1+\mu)} - \frac{c^2g^2}{4m^2g^2(1+\mu)^2}}$$

$$F = \frac{30\sqrt{g}}{\pi}\sqrt{\frac{K}{P(1+\mu)} - \frac{c^2 g}{4P^2(1+\mu)^2}} \quad \text{(8)}$$

Once again simplifying the notation, $a = 1 + \mu$ and $b = c^2g/4$. Equation 8 can then be written

$$F = \frac{187.5}{Pa}\sqrt{KPa - b} \quad \text{(9)}$$

Substituting into this equation the data of the frequency observations,

$$F_1 = \frac{187.5}{P_1 a}\sqrt{KP_1 a - b} \quad \text{(10)}$$

$$F_2 = \frac{187.5}{P_2 a}\sqrt{KP_2 a - b} \quad \text{(11)}$$

All terms except a and b are known. Therefore, solving Equations 10 and 11 simultaneously, first for a and then b,

$$a = \frac{187.5^2 K(P_2 - P_1)}{F_2^2P_2^2 - F_1^2P_1^2} \quad \text{(12)}$$

$$b = KP_1 a - \frac{F_1^2P_1^2a^2}{187.5^2} \quad \text{(13)}$$

Since $b = c^2g/4$,

$$c = 2\sqrt{\frac{b}{g}} \quad \text{(14)}$$

Also since $a = 1 + \mu$

$$\mu = a - 1 \quad \text{(15)}$$

Inasmuch as a and b are based on experiment, the damping constant, c, and the mass ratio, μ, can be determined for any actual setup.

Theoretical considerations developed in this article have been employed in the design of a weighing machine built by the Wright Machinery Co., to whom the author is indebted for permission to use the foregoing findings. Actual performance of the completed machine bore out the theoretical predictions.

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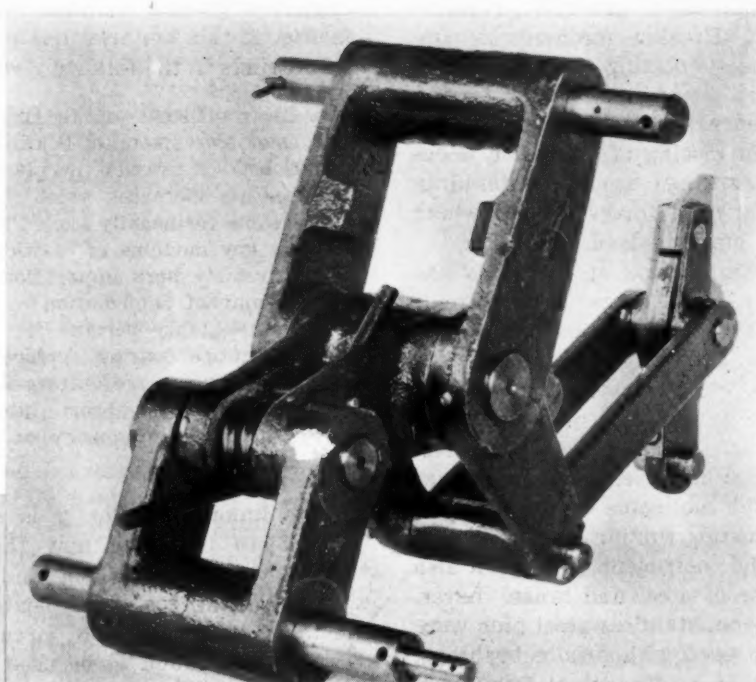


Fig. 1 — Mechanism linkage using bronze pins in cast steel and malleable cast iron

Bearings

for Intermittent, Oscillatory Motion

LUBRICANT is continually carried into most bearings by the rotating action of their shafts. There are other bearing applications, however, such as power circuit breakers, in which no such continual shaft rotation is present to carry lubricant into the bearing. They operate either dry or with grease which may be inserted at assembly or forced in under pressure. Circuit-breaker bearings are unique in that their rotation is confined to a fraction of one revolution in most instances, and operations may be frequent or extremely infrequent.

Despite this limited movement, which eliminates any possibility of carrying lubricant into the bearing, the circuit breaker must stand ready to operate and interrupt a dangerous short circuit at high speed on an instant's notice. To do so, every bearing in the circuit breaker must operate with a minimum of friction regardless of the length of time which has elapsed since the last operation.

Differences in operating requirements for circuit-breaker bearings and those found in most other machines can be illustrated by considering a motor-generator set which develops 50 hp and runs at 3600 rpm. Surface speed of the shaft in the bearings is approximately 35 fps, and the bearings are lubricated by oil and run with pressures in the order of a few hundred pounds per square inch. Surface speeds on typical circuit-breaker shafts or pins, on the other hand, are less than 10 per cent of this motor-shaft speed. But the loads per square inch on these bearings are many times higher, ranging from 2000 to 10,000 psi of projected bearing area, depending upon the particular application. In fact, peak pressures may go considerably higher than 10,000 psi. These high bearing loads are a result of the heavy spring forces used to obtain high-speed acceleration of the breaker contacts. Pin and shaft diameters, moreover, are kept as small as possible to limit the diameter of the friction circles.

Most power circuit-breaker shafts rotate through limited angles when opened; the majority through less than 60 degrees, while few rotate more than 120

This article is an abstract of a paper presented at the Fall Meeting of ASME in Erie, Pa.

degrees. The closing operation generally reverses through the same angle. Breaker mechanisms consist principally of cranks rotating backward and forward as the breakers close and open. During this motion the load generally varies considerably. As an example, during the closing of a breaker, opening springs and contact springs are encountered in a sequence such that the final forces are anywhere from 5 to 25 times the starting load.

Historically, a large combination of materials has been used for circuit-breaker bearings. When circuit breakers were first built for light loads, plain steel pins and steel bearings generally were used. As loads increased, abrasion was encountered, and steel pins and brass bushings were substituted. In some cases it was more convenient to use brass pins directly bearing in steel. *Fig.* 1 is an example of bronze pins in both cast steel and malleable cast iron. With the use of nonferrous metal, rusting and corrosion were greatly reduced. The coefficient of friction also was reduced with the use of steel and brass. Later, in order to reduce corrosion, stainless steel pins were adopted, and these, when used with bronze bushings, again effected a decrease in coefficient of friction.

TEXTOLITE COMPOUND BEARINGS: In the past few years a considerable number of new materials have been studied to determine what materials or combinations of materials might be used for circuit-breaker bearings to obtain even lower coefficients of friction, longer life, higher permissible loadings, or less frequent lubrication. For this intermittent duty where heating was not a limiting factor, a much wider range of materials and combinations of materials could be considered than is possible for continuously rotating bearings. Of the various bearing materials tested, Textolite tubing made of cotton cloth and thermosetting phenolic resin has proved to be far superior to any of the metallic materials. When used with centerless-ground stainless steel shafts, as illustrated in *Fig.* 2, this material has proved superior to metallic bearings in the following ways:

1. Its coefficient of friction is appreciably lower than most metallic bearings; moreover, the coefficient of friction decreases as the load on the bearing increases which makes it particularly desirable for heavily loaded bearings
2. Its low modulus of elasticity enables it to absorb extremely high impact loads without cracking or permanent deformation
3. Its flexibility permits it to distribute stress over the entire bearing surface even with minor misalignments in the bearing seats
4. Its ability to absorb lubricant into its surface reduces the frequency of maintenance for lubrication.

The linkage of *Fig.* 2 is shown schematically in *Fig.* 3 to illustrate how the operating forces act through the mechanism and bearings.

The advantages which Textolite bearings offer over metallic bearings for circuit-breaker applications have caused them to be used widely during the past few years for all locations except trip latches or other locations where very low torques are available. Ball, roller, or needle bearings have been used for these locations because of their lower coefficient of friction at lighter loads. Textolite bearings are normally held in place by a press fit. As Textolite is a compressible material, considerably more interference between bore and bushing can be tolerated than with metallic bearings. A Textolite bearing as much as 0.020-in./in. of bearing diameter larger than the hole in the supporting member may be pressed into place without damage to the bearing, providing sufficient pressure is available. As it is pressed into place, however, the compression decreases the internal diameter of the bearing approximately the same amount as the in-

Fig. 2—Cranks and rollers of an air mechanism showing use of Textolite bushings and stainless steel pins

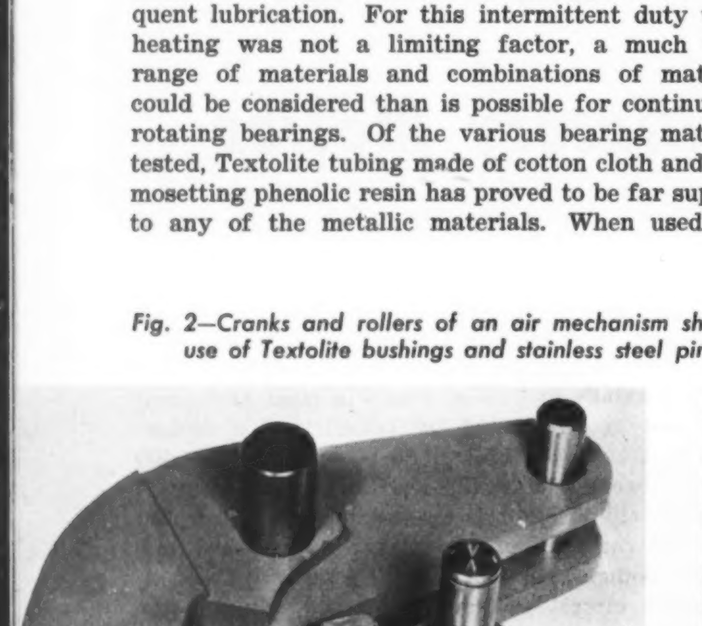
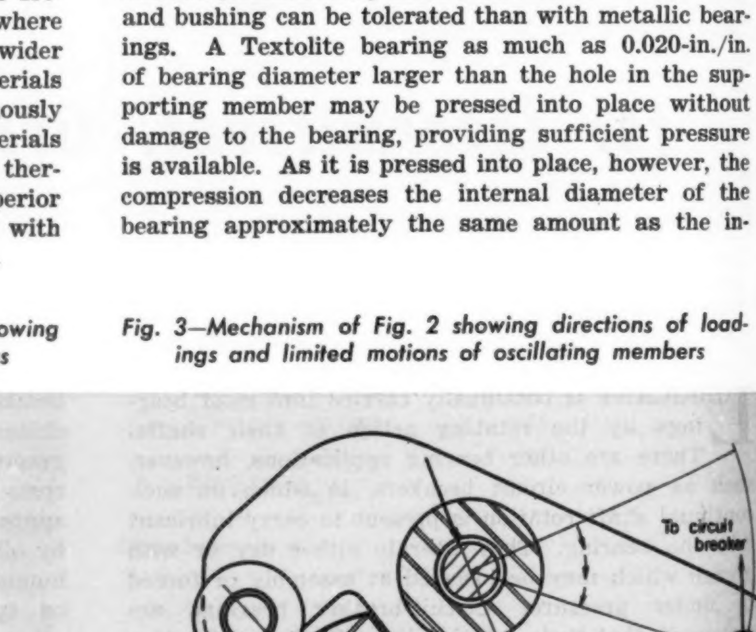


Fig. 3—Mechanism of Fig. 2 showing directions of loadings and limited motions of oscillating members



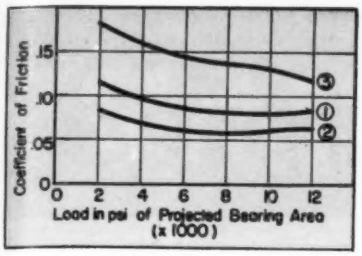


Fig. 4 — Friction-load curves for stainless steel pins in Textolite bushings. (1)—Bushing interference fit, 0.003-0.006-in.; (2)—0.008-0.011-in.; (3)—0.013-0.016-in.

terference between its outside diameter and the hole in the supporting member. Consequently the amount of press fit between the outside of the bearing and the hole in the supporting member must be held to a fairly close tolerance in order to insure uniform internal diameter in the bearing when in place. A clearance of approximately 0.007-in. is desirable between the internal diameter of the bearing and the shaft to provide for shaft deflection between the two ends of the bearing and also to permit slight expansion during periods of high humidity. Wall thickness is 1/8-in. for small and medium-sized bearings, while a 3/16-in. is used on larger ones.

Press Fit Controls Friction

The coefficient of friction between shaft and bearing depends upon the amount of press fit used when inserting the bearing in the supporting member. *Fig.* 4 shows curves taken on 3/4-in. bearings with three different amounts of press fit, and with the same grade of lubricant in each case. Where only 0.003 to 0.006-in. was used, it was found that the shaft sank into the bearing and made contact with considerably more than 120 degrees of its surface. With 0.008 to 0.011-in. press fit, the Textolite was considerably more compressed, thus giving better support, and contact was made over 90 to 120 degrees of surface, which decreased the coefficient of friction appreciably for all loads. With 0.013 to 0.016-in. press fit, less than 90 degrees of contact were obtained and the coefficient of friction was almost doubled. This substantiates a large number of tests indicating that 0.008 to 0.011-in. press fit per inch of bearing diameter provides the optimum operating conditions for circuit-breaker bearings.

Fig. 5 shows the method of installing these bearings. The hole in the supporting member should of course be smooth with a small chamfer on the side from which the bearing is to be pressed into place. This is necessary to prevent damage to the outer surface of the bearing while it is being pressed into place. A small bevel at the ends of the outer surface of the bearing has also been found helpful in eliminating shearing damage during the pressing operation. In most cases the friction of the press fit will hold the bearing in place. If very short bearings are used, or if the impact loading is especially severe, it may be necessary to install retaining plates at either end of the bearing.

Life tests of actual circuit-breaker bearings show also that considerably higher bearing loads per square inch of projected area are permissible with Textolite bearings than with metallic bearings. Whereas 7000 to 8000 psi of projected area is considered maximum for most metallic bearings, 10,000 to 12,000 psi may be used without difficulty on Textolite bearings. This permits the use of smaller shaft sizes, thereby resulting in further reduction in friction and faster breaker operating times. Textolite bearings may also be subjected to impact loads which in a few operations would elongate metallic bearings beyond further possible use. This is particularly desirable in circuit-breaker applications, as it eliminates the necessity of dashpots or other buffers in many locations where they are installed primarily for protection of the bearings.

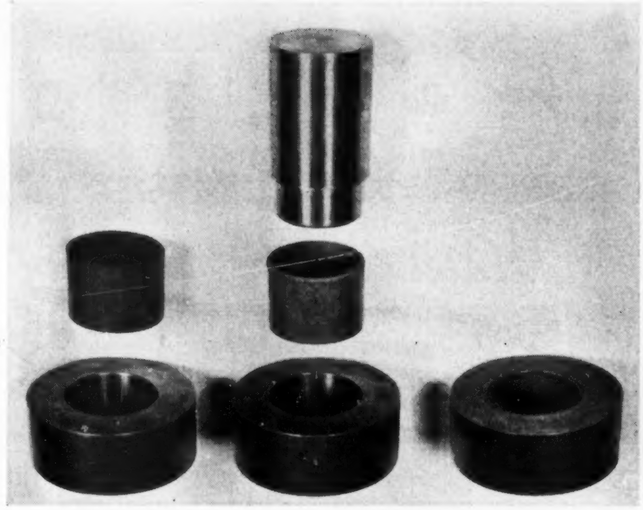


Fig. 5—Steps in assembly of a Textolite bushing into a hole in the supporting member

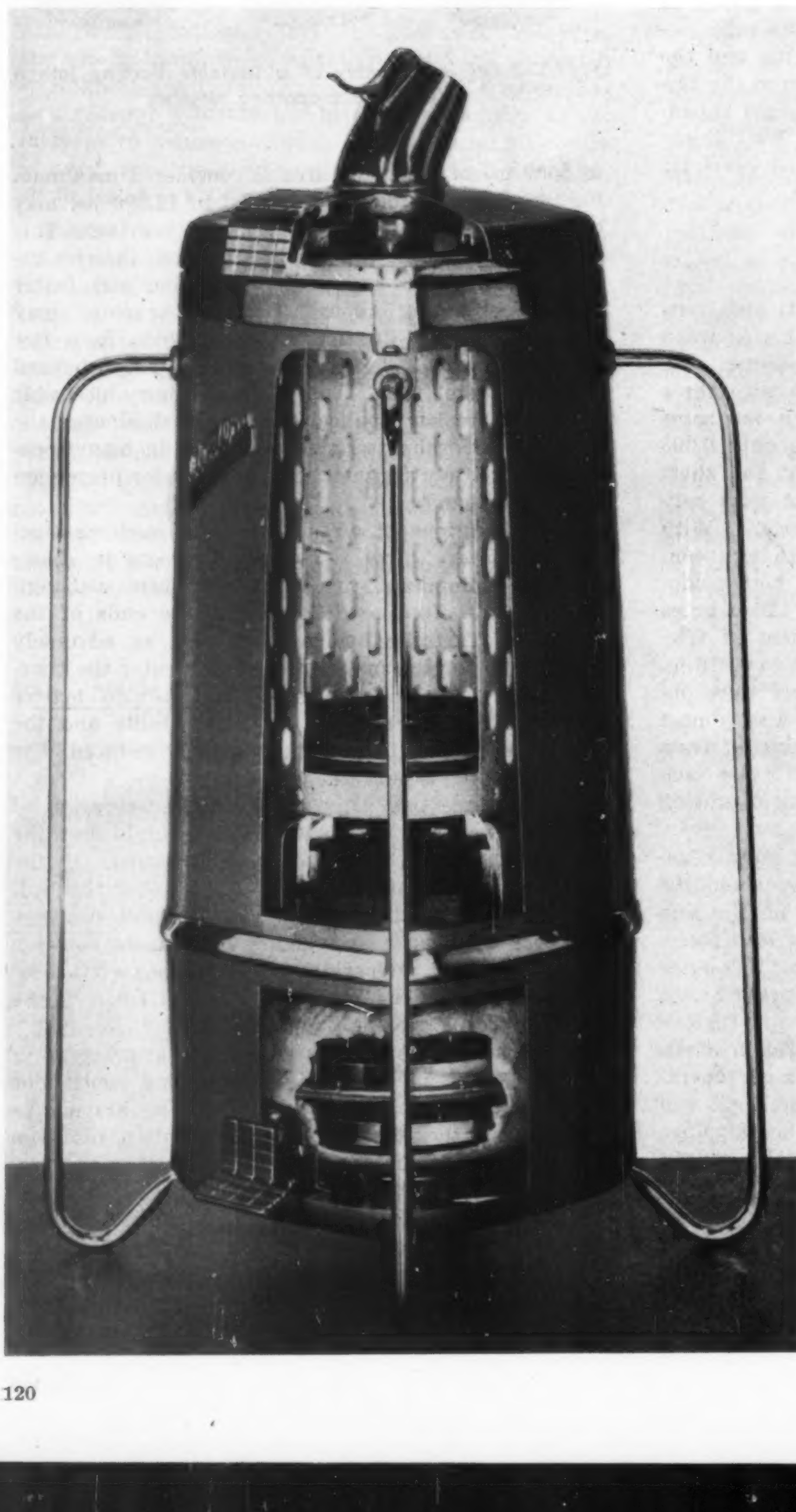
Many locations of circuit breakers, such as steel mills, are very dirty. The use of grease in sleeve bearings minimizes the effect of dust although washers are often used to protect the ends of the bearings. Textolite bearings are not as adversely affected by foreign matter which may enter the bearing as are metallic bearings. Any foreign matter is embedded rather easily in the Textolite and the scoring action on the shafts is greatly reduced. The bearing itself is not affected.

Temperature has little effect on the operation of Textolite bearings unless it rises so high that the strength of the cotton fibers is impaired. If the temperature is raised higher, actual charring will take place. This limits the use of Textolite bearings in most applications involving continuous rotation where the local temperature at the inner surface of the bearing may be quite high, unless some means of keeping the bearing cool is provided.

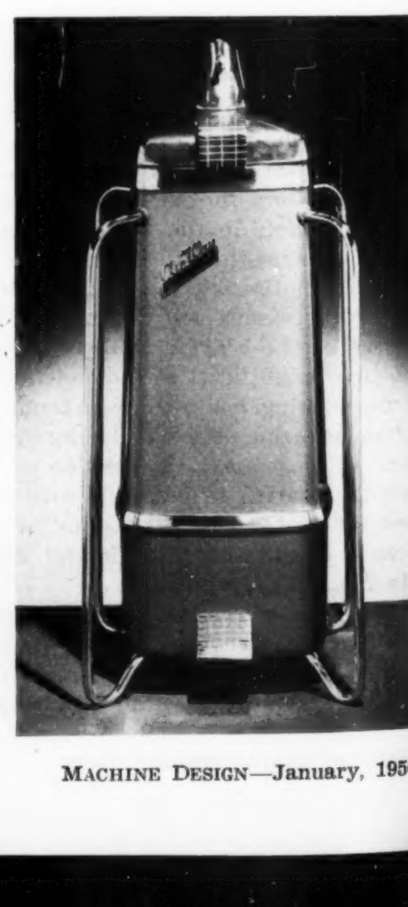
Lubrication has always presented a problem in circuit breakers. Without complete and continuous rotation little grease is drawn into the bearing as it operates, therefore the only lubrication that can be depended upon is that which is inserted at assembly. As has been mentioned, Textolite bearings are made from a cloth embedded in phenolic resin, and the spaces in the cloth are seldom entirely filled with the resin on the inner surfaces. Thus, the little pockets which are left retain grease much more readily than smooth metal-finished surfaces.

CONTEMPORARY

Cleaner Features Disposable Filter



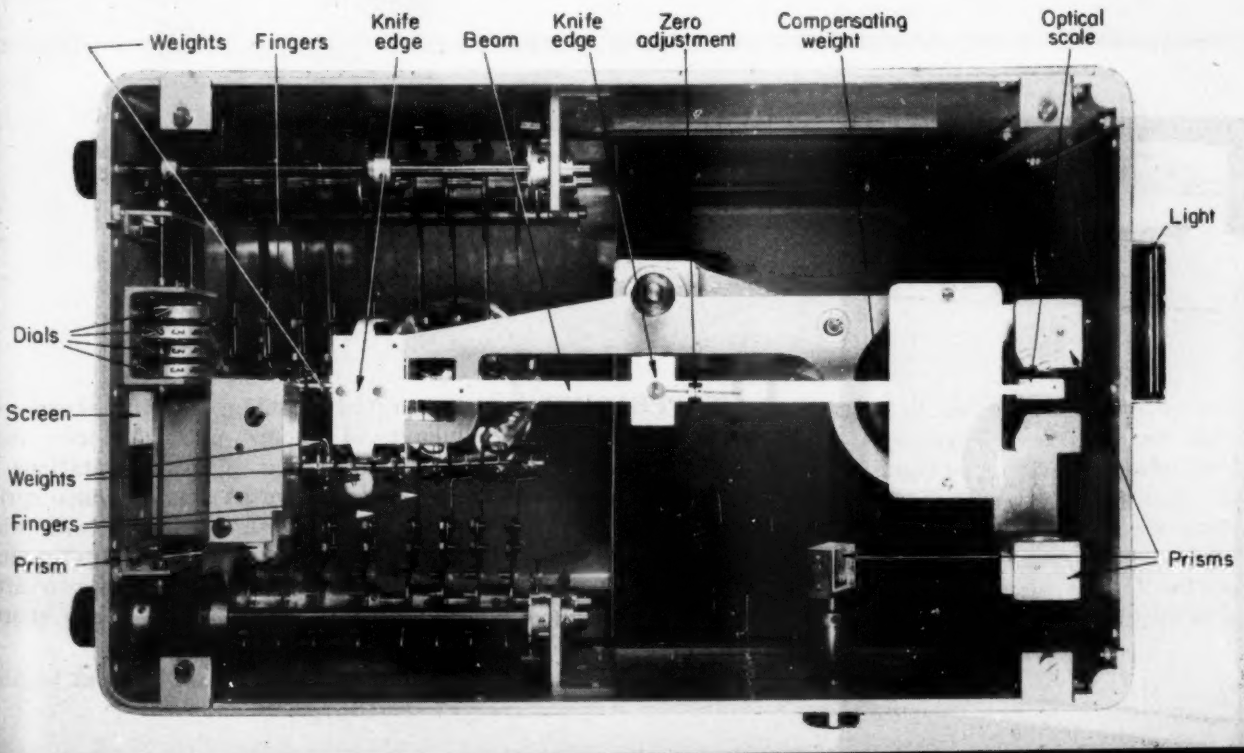
MODERN styling of the tank type vacuum cleaner, left and below, serves in a functional as well as ornamental capacity. Vertical chromed bars dress up the appearance and are used as gliders in moving the unit from one room to another. The simple design of the cleaner permits its operation in either vertical or horizontal position, a swivel intake connection allowing the power unit to remain stationary while cleaning an area of 500 square feet. A disposable paper filter bag eliminates the objectionable task of cleaning out the dust collecting compartment, while an insulating jacket around the motor and ball bearing mounting of all rotating parts contribute to quiet operation. Manufacturer: Air-Way Branches Inc., Toledo, O.

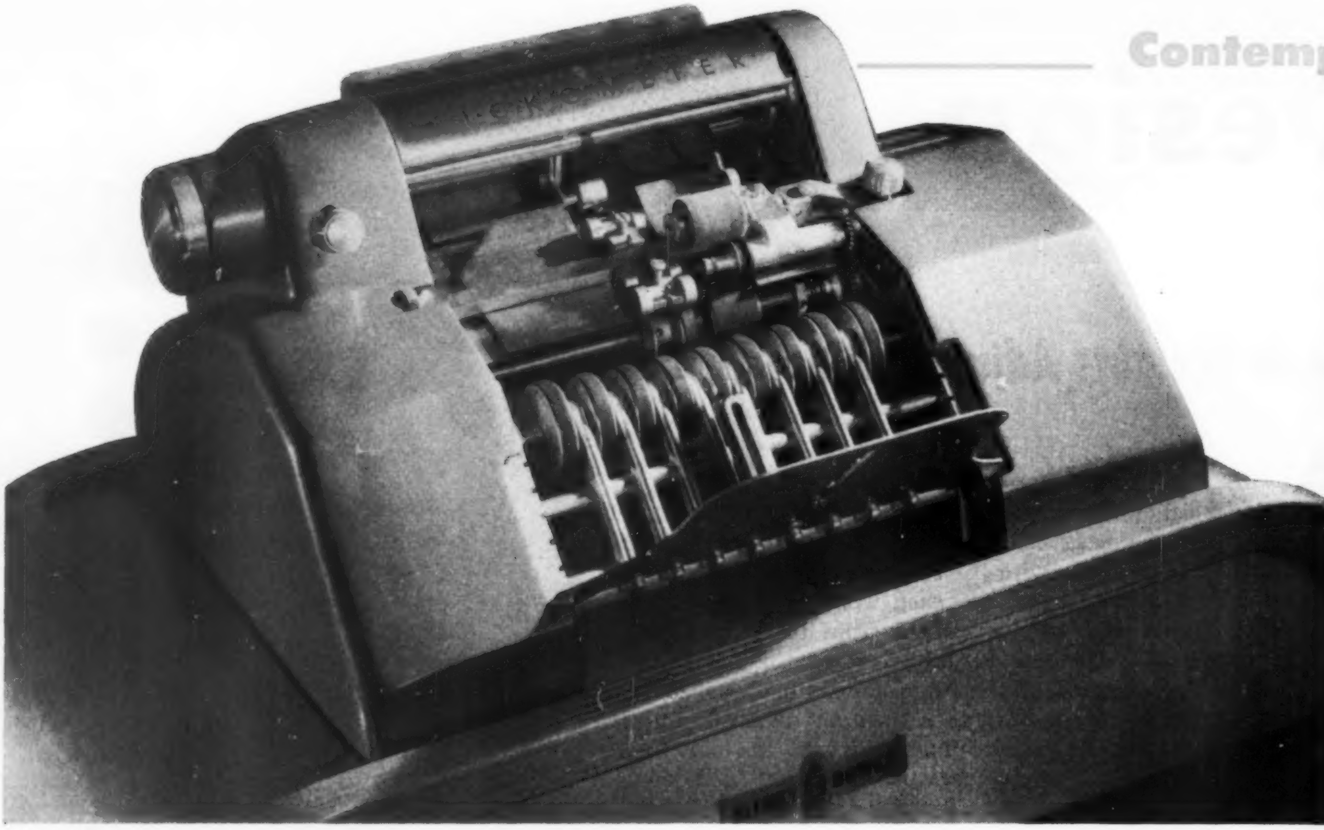


Design

Scale Weights Adjusted Mechanically

A DISTINCT innovation in the precision weighing field, the balance shown, right, employs mechanical fingers to manipulate the weights. The one-pan, two-knife edge unit makes all weighings with the beam under load, weights being removed instead of added when weighing an object. Pan and weights are both suspended from the same end of the beam, as shown in the top view of the balance, below. The opposite end of the beam carries a conpensating weight which also serves as a dashpot or air-damping device, a series of plates equal in area to the pan to eliminate errors due to temperature changes, and an optical scale. A set of prisms directs a light through the optical scale and projects it onto a ground glass scale on the front of the balance. To weigh a sample, knobs on the front of the instrument are turned which, through a series of cams and fingers, remove weights from the beam. The total of the weights removed registers automatically on a set of dials, the last three decimal places in the total weight figure being read from the projected scale. Since the total weight on the left end of the beam is always the same, 200 grams, the balance has a constant sensitivity throughout its capacity range. Manufacturer: E. Mettler, Switzerland; distributer, Fisher Scientific Co., Pittsburgh, Pa.





Cards Counted Automatically

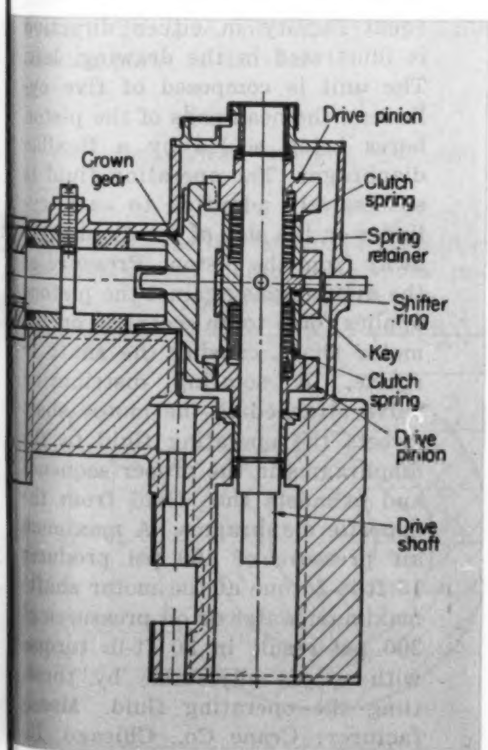
ACCURATE counting and imprinting of 900 2 by 2-inch cards per minute is automatically accomplished with the Tickometer machine shown in the illustration, above. Material to 5 by $8\frac{1}{2}$ inches in size and varying from 0.003 to 0.016-inch in thickness is automatically drawn from a feed hopper by rubber rollers at the rate of 30 inches per second. These rubber rollers driving against steel separator bars carry the material to the trip mechanism which actuates two counters, one a cumulative total and the other a setback type counter. After passing the trip mechanism and after date, code number, endorsement or canceling markings have been printed on the pieces they are automatically stacked. At the point in the feeding operation where the cards are separated, a throw-out shoe which is adjusted to the thickness of the items being handled prevents more than one piece at a time from passing the separator. Card thickness variations of a few thousandths can be detected. Mfg.: Pitney-Bowes Inc., Stamford, Conn.

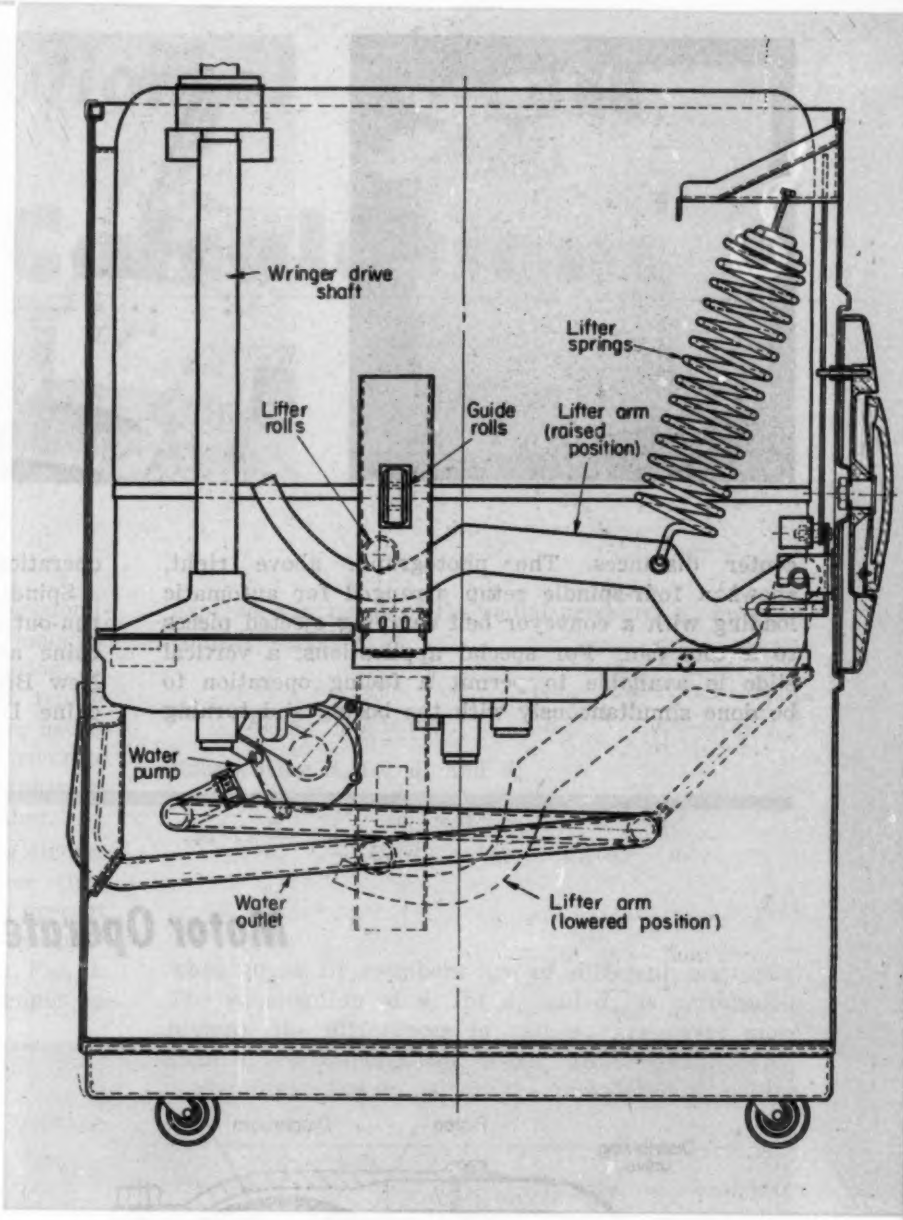
Washer Uses Constant-Force Linkage

DISAPPEARING wringer on the washing machine shown, left, requires an interesting linkage to impart uniform force during elevation from the stored position. Because of space limitations, tub and wringer assemblies move up and down as a unit. Since springs are used in lifting, it was necessary to design a lifting lever whose effective length would shorten as the unit moves upward. The drawing, top next page, shows the developed curves on the two lifting arms that result in a constant lifting force as the springs shorten.

Rolls support the washer unit on the arms and allow the point of contact to change throughout the lift. Four guide rolls, two on each side of the basket, locate the basket in the cabinet while permitting vertical travel.

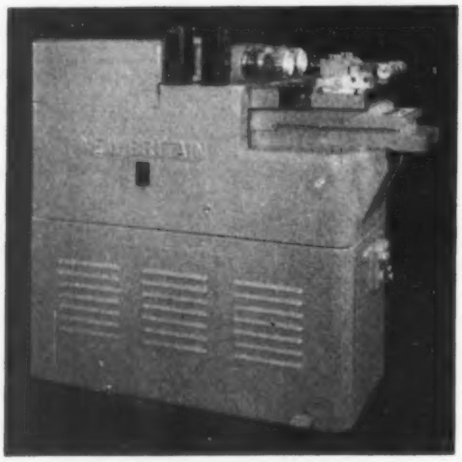
The wringer head drive, shown in the drawing below, employs spring clutches to transmit power to the rolls. A sleeve-like shifter ring mounted around the spring retainer can be moved either up or down by the shift lever. When the washer motor is operating, shaft, spring retainer and shifter pin or key are rotating. Moving the shifter ring engages the key with one of the clutch springs. This spring expands radially, contacting the inside of the retainer. Since the springs are fastened to the drive pinions, power is transmitted from the retainer through the spring and to the pinion. Most of the load is carried through the spring, the key carrying only a small part of the load and is therefore easily disengaged. Reverse wringer rotation is accomplished through the other pinion in the same manner. Manufacturer: Automatic Washer Co., Newton, Iowa.

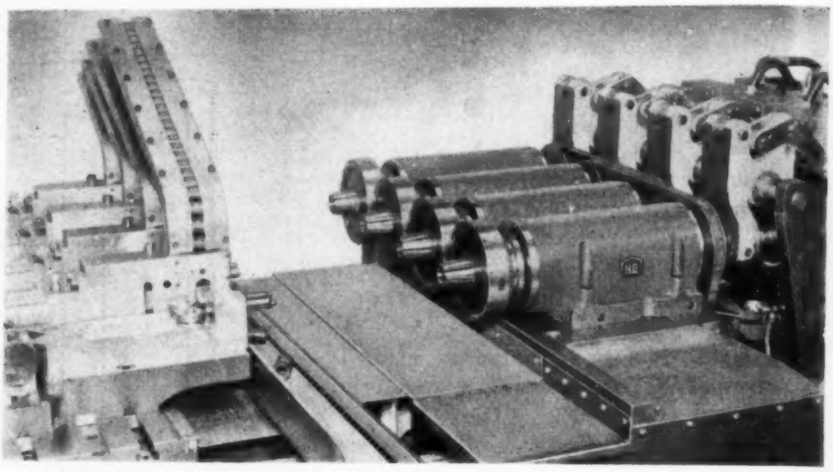




Slides Permit Contour Boring or Turning

VERSATILITY is the keynote of the precision boring machine shown at the upper left of next page which combines the compound motion of two slides to accomplish contour boring or turning in addition to making straight turning or facing cuts. Longitudinal and contour slides are positively actuated by cams opposed by pneumatic cylinders. Added flexibility is attained from the novel spindle arrangement—from one to four V-belt driven spindles can be mounted on the machine at varying heights and

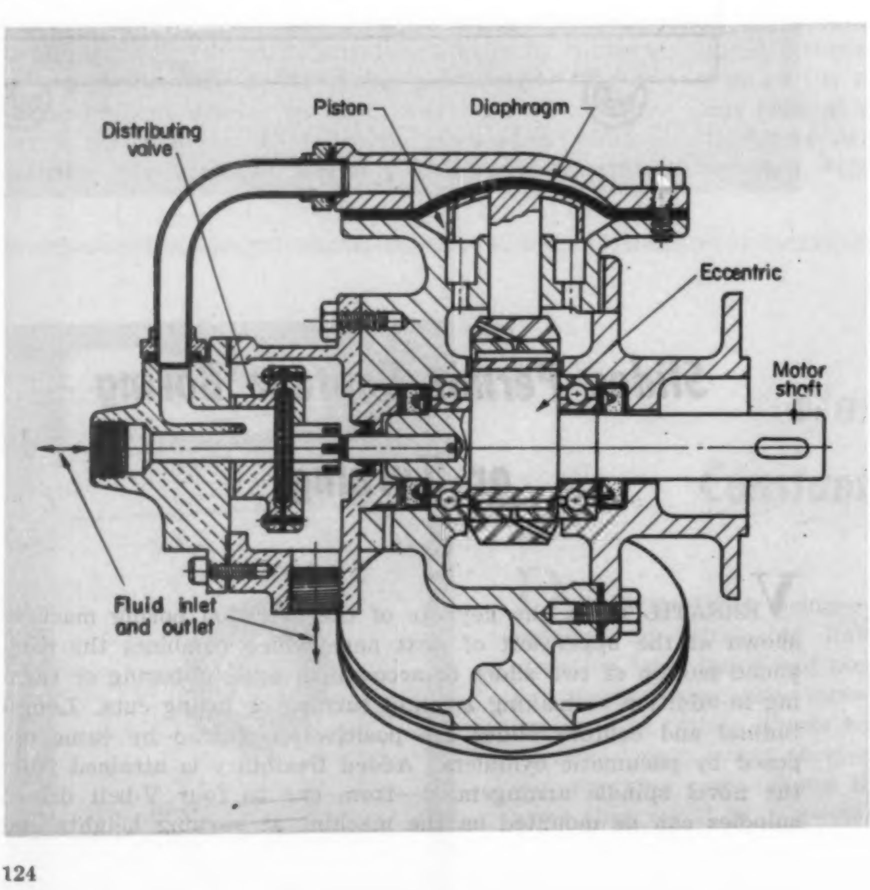




center distances. The photograph, above right, shows a four-spindle setup arranged for automatic loading with a conveyor belt carrying ejected pieces to a tote pan. For special applications, a vertical slide is available to permit a facing operation to be done simultaneously with the boring and turning operations.

Spindles, operating to 7500 rpm, have maximum run-out of 0.000025-inch which contributes to the machine accuracy of 0.0001-inch. Manufacturer: The New Britain Machine Co., New Britain-Gridley Machine Div., New Britain, Conn.

Motor Operates On All Fluids



A FLUID motor that operates on air, water or oil pressure with equal facility in either direction is illustrated in the drawing, left. The unit is composed of five cylinders, the head ends of the piston bores being sealed by a flexible diaphragm. The operating fluid is successively admitted to each cylinder on the side of the diaphragm away from the piston. Pressure of the diaphragms against the pistons applies force to an eccentric on the motor shaft, causing the shaft to rotate. A rotating distributing valve attached to the motor shaft directs the operating fluid to the diaphragms in the proper sequence and exhausts the fluid from the opposite diaphragms. A maximum air pressure of 100 psi produces 10 ft-lb torque at the motor shaft; maximum water or oil pressures of 300 psi result in 30 ft-lb torque, with speeds adjustable by throttling the operating fluid. Manufacturer: Crane Co., Chicago, Ill.

Graphical Solution of

Press Fit Calculations

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PRESS or force fits of machine parts are economical for many types of work in the machine design field. Mathematical determination of radial pressure between hub and shaft, hub tensile stress, and required force to make the fit are usually tedious and time-consuming. Sufficiently accurate solutions for these quantities can be made graphically with a considerable saving in time and labor.

PRESS FIT ANALYSIS: Members of a press fit are usually considered as thick cylinders. Since they are open at the ends, no direct axial stress is present and Birnie's equations for open cylinders are applicable. In the compound cylinders shown in *Fig.* 1, pressure at contact surfaces will cause changes in the diameters which may be expressed as

$$\Delta d_k = d_k - d_c \text{ and } \Delta d_m = d_c - d_m$$

Total allowance, or intentional difference between mating parts, is therefore

$$\Delta d_k + \Delta d_m = d_k - d_m = A$$

The diametrical changes, Δd_k and Δd_m, may be expressed in terms of the radial pressure, p_c, between contact surfaces as

$$A = \frac{p_c d_k}{E_k}\left(\frac{d_k^2 + d_i^2 - c_k}{d_k^2 - d_i^2}\right) + \frac{p_c d_m}{E_m}\left(\frac{d_o^2 + d_m^2 + c_m}{d_o^2 - d_m^2}\right)$$

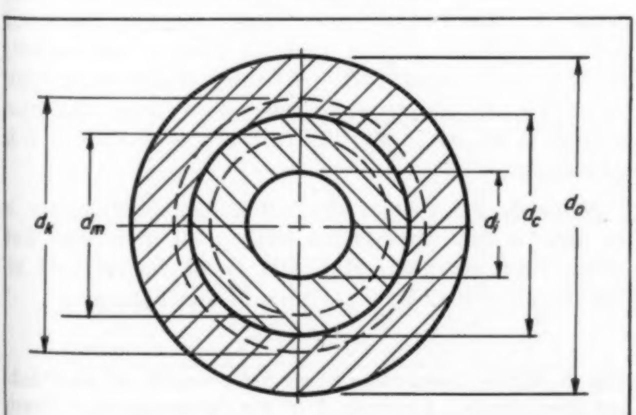
Substituting d_c for d_k and d_m

$$p_c = \frac{A}{d_c} \div \left[\frac{d_c^2 + d_i^2}{E_k(d_c^2 - d_i^2)} + \frac{d_o^2 + d_c^2}{E_m(d_o^2 - d_c^2)} - \frac{c_k}{E_k} + \frac{c_m}{E_m}\right] \quad \text{(1)}$$

when press fit members are of different materials. The substitution of d_c for d_k and d_m is permissible because the differences in values are never more than a few thousandths of an inch.

For press fits in which the members are of the

Fig. 1—Cross section of typical press-fit member showing dimensions used in evaluating radial pressure, assembly force and hub stress



Nomenclature

A = Allowance or intentional difference in size of mating parts, inches
c = Poisson's ratio of lateral contraction
d = Nominal shaft diameter, inches
d_c = Shaft outside diameter, inches
d_i = Shaft inside diameter, inches
d_k = Expanded hub inside diameter, inches
d_m = Compressed shaft outside diameter, inches
d_o and D_o = Hub outside diameter, inches
D_i = Hub inside diameter before fit is made, inches
E = Modulus of elasticity of shaft or hub material, psi
F = Assembly force, lb
f = Coefficient of friction
L = Hub length, inches
k = Subscript denoting one material
m = Subscript denoting second material
p_c = Radial pressure between hub and shaft, psi
s_{ct} = Hub tensile stress, psi

same material, Equation 1 simplifies to

$$p_c = \frac{AE\,(d_c^2 - d_i^2)\,(d_o^2 - d_c^2)}{2d_c^3\,(d_o^2 - d_i^2)} \quad \text{.......... (2)}$$

When the inner member or shaft is solid, $d_i = 0$, and Equations 1 and 2 are easier to evaluate.

The tensile stress at the inner surface of the outer member or hub is

$$s_{tc} = p_c \left(\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2} + c \right) \quad \text{.......... (3)}$$

This relationship is usually used for cast iron hubs and a simpler one for steel hubs. For the latter,

$$s_{tc} = 3A\ (10^7) \quad \text{.......... (4)}$$

Experience has shown that Equation 4 is sufficiently accurate for design of steel hubs for all ordinary press fits.

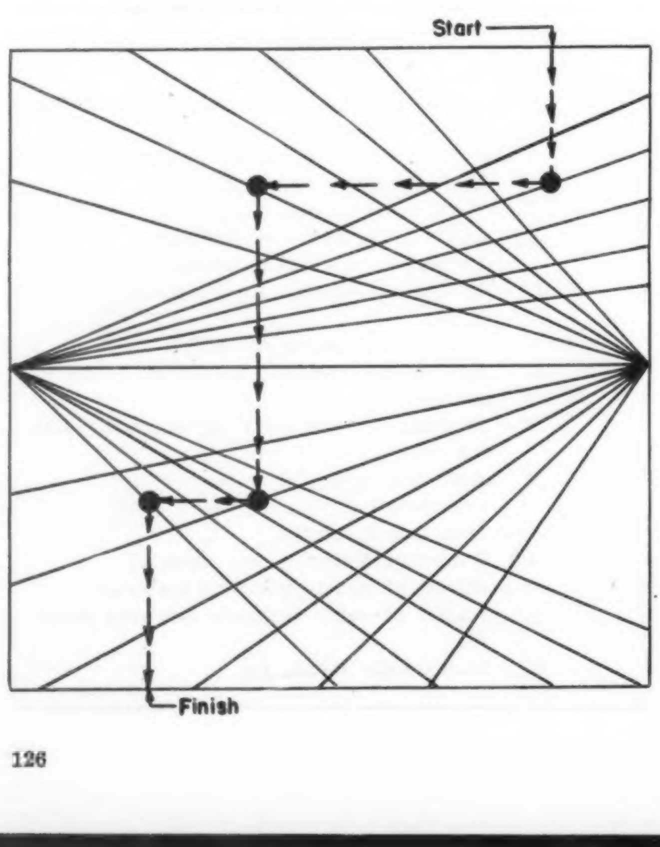
The axial force required to assemble a press fit is a function of hub length and thickness, friction between hub and shaft, and diameter ratio of the members. In equation form,

$$F = \pi dLfp_c \quad \text{.......... (5)}$$

This relationship takes into consideration the factors affecting the required force, and all except the coefficient of friction can be obtained by measurement of the members.

Coefficients of friction for press fits vary widely, 0.04 to 0.25 being the ordinary range. Values of 0.10 to 0.13 are commonly used in computing the axial force required for assembly. In cases where the shaft is rough or the hub bore uneven, higher values, 0.20 to 0.25, will give more accurate results. Lower values, 0.05 to 0.075 are used for determining holding power of the press fit when hub or shaft is axially loaded. Lubrication of either member before making the press fit has little effect on the coefficient of friction because the lubricant is usually squeezed out during assembly.

Fig. 2—Operating procedure for Figs. 3 and 4. Following arrowed line determines required assembly force and radial pressure between hub and shaft



Torsional loading of hub or shaft may cause slippage. Therefore, the coefficient of friction used to determine the allowable torque is usually taken as 0.10. Allowable torque is found from the relationship.

$$T = 1/2\pi f p_c L d^2 \quad \text{.......... (6)}$$

Torques greater than the allowable will generally cause local slippage near the end of the fit. With alternating torques, stress concentration and rubbing corrosion at hub face may cause fatigue failure. In general, the maximum torsional holding power should be used only for static torque applications.

Graphical Solutions: As can readily be seen, mathematical determinations of radial pressures, hub stresses, and required assembly forces are cumbersome. To simplify these calculations the intersection charts shown in *Figs.* 3 and 4 and the stress chart in *Fig.* 5 have been constructed. These charts are valid for solid shafts and for hollow shafts in which the inside diameter is not more than 25 per cent of the outside diameter. Where stresses are above the yield point of the material these charts cannot be used because plastic flow will affect the values. Two typical problems will be solved to illustrate use of the charts.

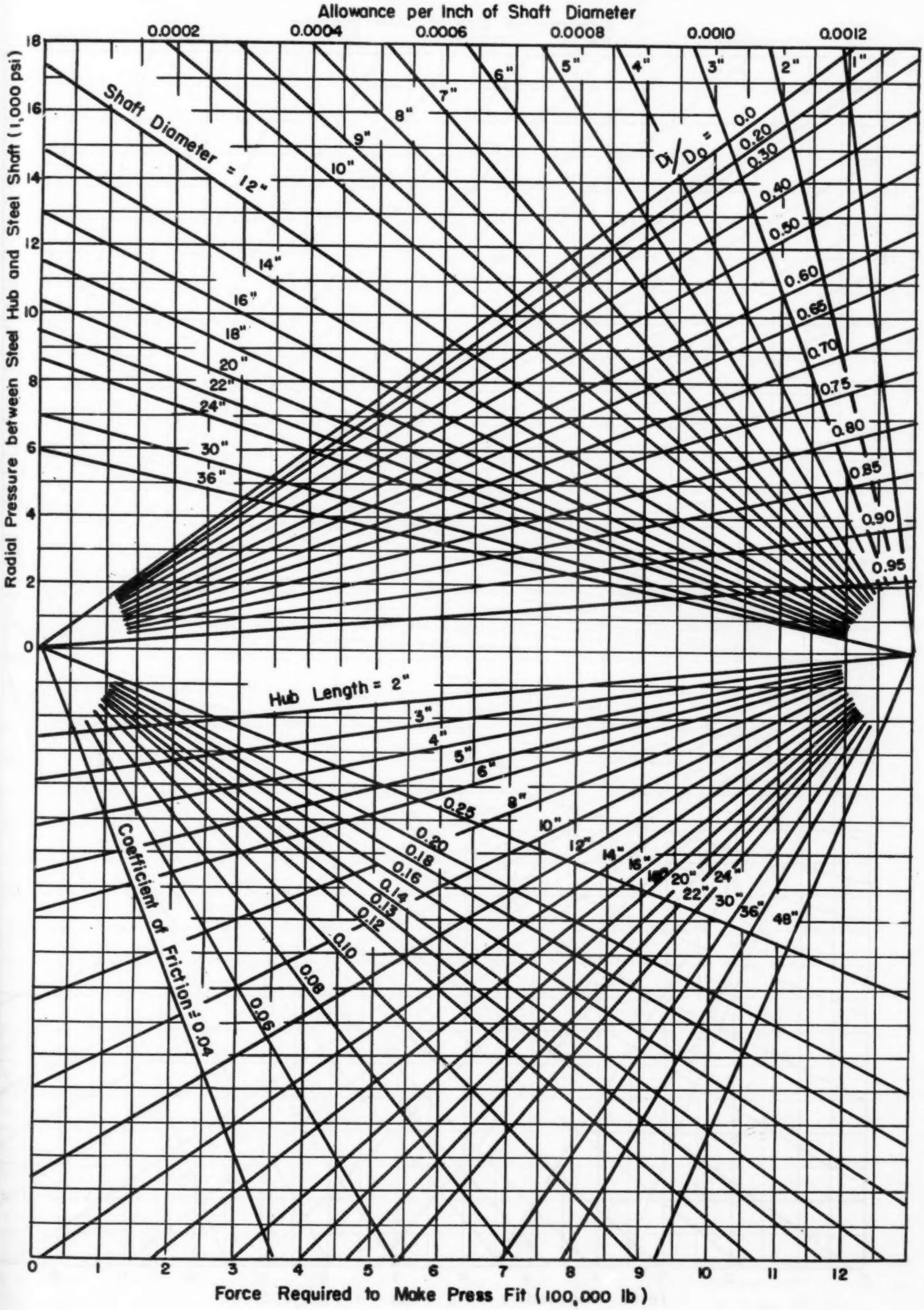
Example 1: A steel shaft 10 inches in diameter is to have a steel hub 14 inches long pressed on its end. Hub outside diameter is 20 inches. An allowance of 0.0008-inch per inch of shaft diameter has been specified. Find the radial pressure between contact surfaces, hub stress, and assembly force required.

Fig. 3 is for steel shafts and hubs. Ratio of shaft to hub diameter is 10/20 = 0.50. Enter *Fig.* 3 at 0.0008-inch allowance per inch of shaft diameter and project vertically downward to the 0.50 diameter-ratio curve. From here run across horizontally to the 10-inch shaft diameter curve and read the hub radial pressure as about 9000 psi. Then project vertically downward from the 10-inch shaft diameter curve to the 14-inch hub length curve. Assuming a coefficient of friction of 0.10, project horizontally to this curve and then vertically down to 400,000 lb, which is the required assembly force. Solution outline is shown in *Fig.* 2. Hub stress from *Fig.* 5 is 24,000 psi.

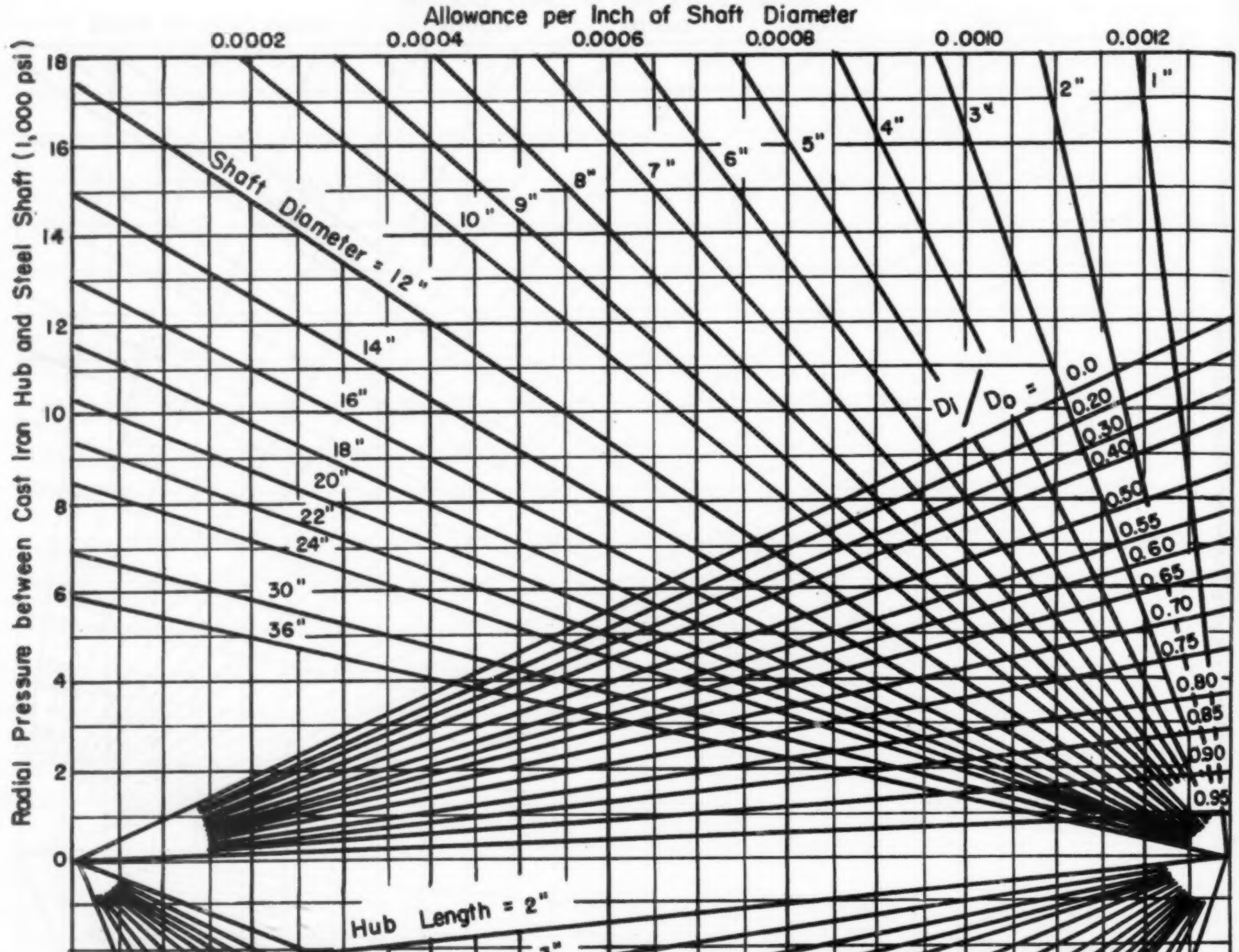
Example 2: A steel shaft 9 inches in diameter is to have a cast iron crank with a 10 inch long hub pressed on its end. Hub OD is 12 inches and allowance is 0.0012-inch per inch of shaft diameter.

Fig. 3—Right—Intersection chart for press fits of steel hubs on steel shafts. Example 1 in the foregoing text clearly illustrates use of this chart

STEEL HUBS ON STEEL SHAFTS



CAST IRON HUBS ON STEEL SHAFTS



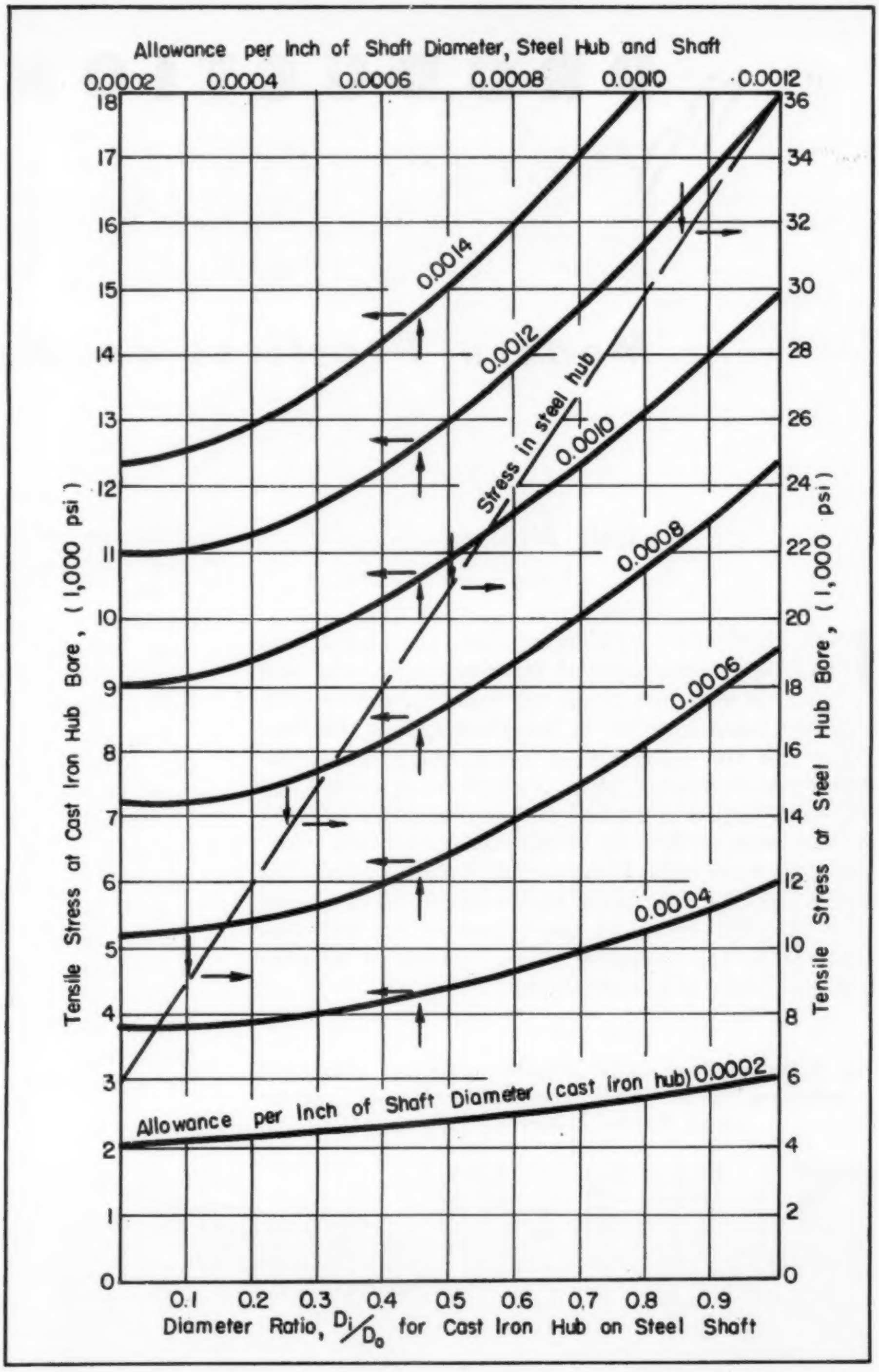


Fig. 4—Left — Intersection chart for press fits of cast iron hubs on steel shafts. Example 2 in the text illustrates use of this chart

Fig. 5—Right—Hub stresses for cast iron and steel hubs on steel shafts. When entering chart, use diameter ratio for cast iron hubs and allowance for steel hubs

Fig. 4 is for steel shafts and cast iron hubs. Enter *Fig.* 4 at an allowance of 0.0012-inch per inch of shaft diameter and project down to the diameter ratio of 9/12 = 0.75. From here run across horizontally to the 9-inch shaft diameter curve and read the radial pressure as 4300 psi. Next, project vertically down to the 10-inch hub length curve, across horizontally to the 0.10 friction factor curve and down to the assembly force of 125,000 lb. Solution outline is similar to *Fig.* 2. Hub stress from *Fig.* 5 is determined as 15,250 psi.

CHART APPLICATIONS: These charts find many applications in office and shop. Besides determination of radial pressures, hub stresses, and assembly forces, any other variable may be solved for if the dimensions of the members are known. One particular use which has been found a great time-saver is determination of coefficients of friction. When a new type of press fit is put into production the approximate required assembly force is found graphically from the appropriate chart, using an assumed coefficient of friction. The members are then assembled and an accurate value of the actual assembly force required is obtained from instruments on the press. Entering the chart with the actual assembly force and dimensions of the members, the actual coefficient of friction is found. Several tests of this type on similar members permits determination of an average value of the friction coefficient which can be used for all future work on these members.

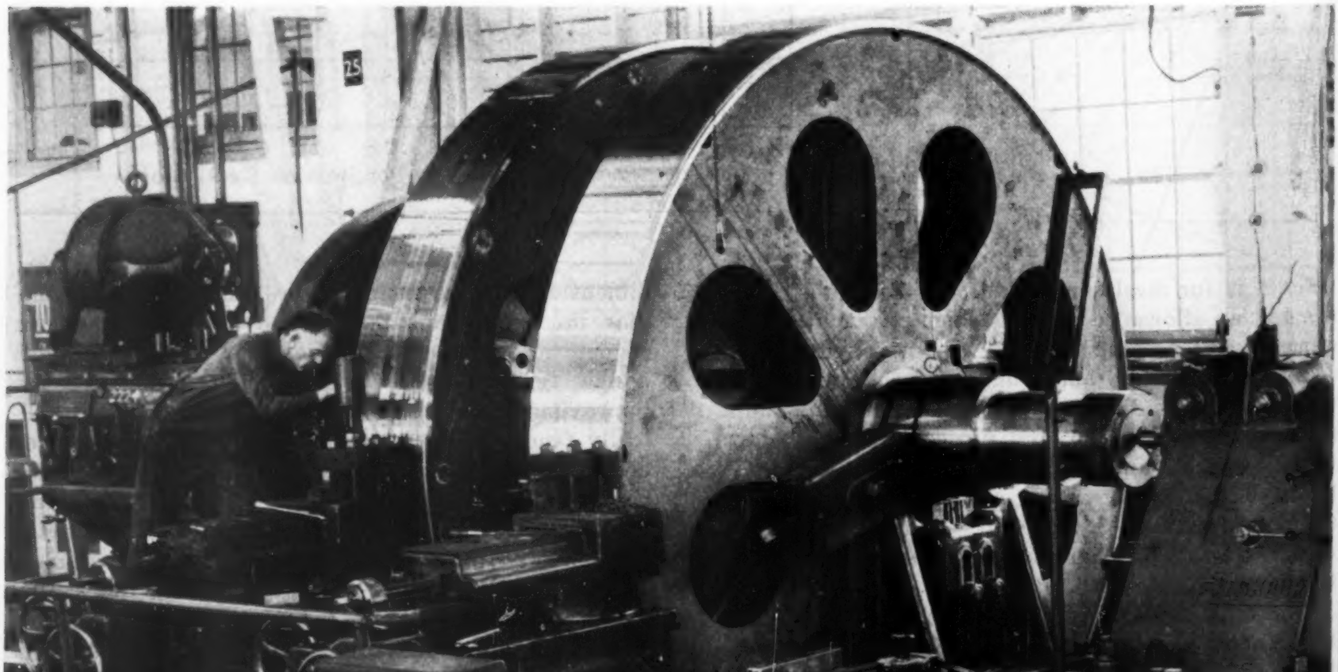
PRODUCTION AND DESIGN

Modern Practices in Manufacture

Big-Scale Accuracy

IN A MODERN turbine-driven ship the biggest and heaviest single unit of its machinery, the bull gear, is also just about the most microscopically accurate thing aboard. Although weight of the two gears and the shaft shown being machined at the Sunnyvale, California, plant of the Westinghouse Electric Corporation is 51,200 lb, tolerances for the operation must be held to as close as 0.0003-inch. The two wheels are welded together on a shaft to form the final stage of a gear train that reduces the 6000 rpm of a 7500-hp turbine to the approximately 100 rpm required in the propeller drive. Diameter of each bull-gear wheel is 146.392 inches.

Fig. 1—Below—Final stage of lathe operation. Machinist is inscribing "target patch" line 0.003-inch wide used for lining up the helices in hobbing



The two bull-gear wheels have opposite helix angles, each wheel containing 693 teeth with a diametral pitch of five. The two helices must intersect at an hypothetical line exactly midway between the two inner rims. Hubs of the gear are forged steel, sides are of steel plate, and the rims and shaft are high-carbon steel forgings.

The wheels are turned on a big lathe as shown in *Fig.* 1 in four major operations: Wheel diameters, faces of rims, beveled corners, and wheel shoulders or "dowels." Accuracy surpasses that found in a fine watch. For any of these surfaces, the maximum "indicator runout"—the amount that dimensions may vary from the computed figures—is less than three ten-thousandths of an inch. The shaft is supported on steadyrests and lathe centers.

Great care is taken to make sure that the big wheels and their shaft are properly aligned before machining. The shaft is turned true to ensure that the wheels will turn in the exact vertical plane of the

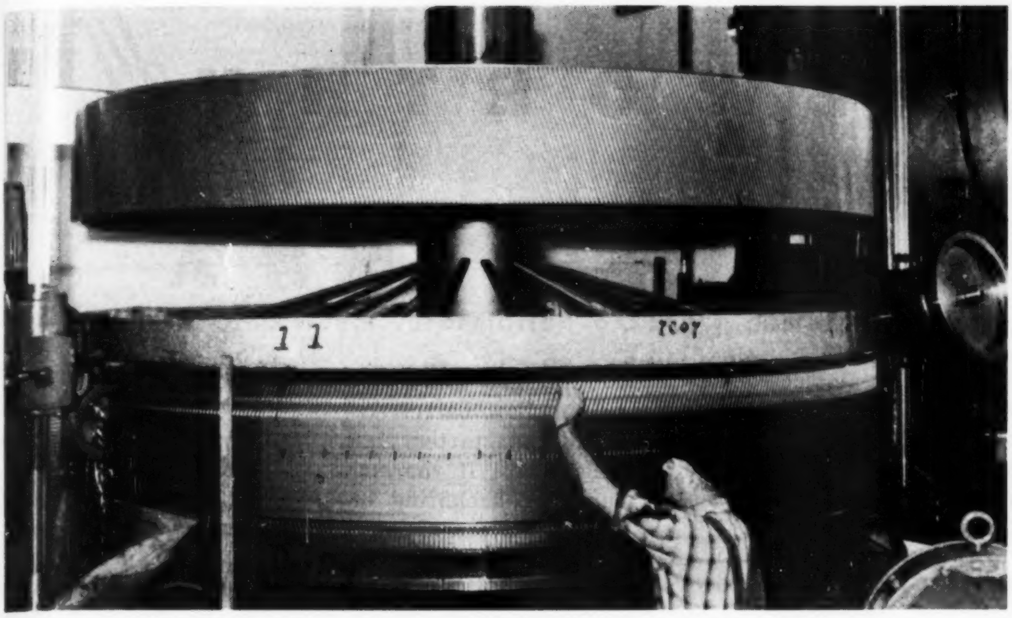


Fig. 2 — Left — Hobbing of large bull gears for ship propulsion. Teeth are completed on the upper wheel and hobbing has just been started on the lower wheel

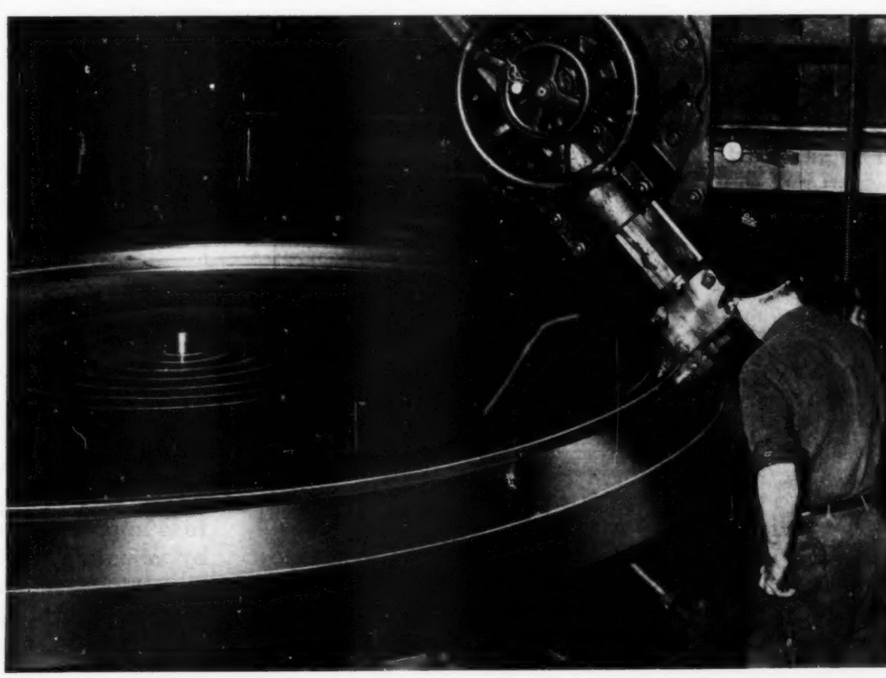
shaft journals. Prior to the lathe machining, the wheel has been completely welded, normalized, and shrunk on its shaft.

Hobbing of the teeth had been completed on one wheel and is just starting on the second in *Fig.* 2. Square "target patches" are used on both wheel rims and can be seen just to the right of the operator. A thin locating line runs through these patches and provides reference points for lining up the helices.

Hobbing is done by a rotary multipointed tool not unlike a pine cone in appearance at the extreme left of the lower wheel. Twelve minutes are required for the wheel to revolve and with each revolution the hob advances into fresh metal. Both a rough cut and a finish cut are made for each wheel. A rough cut takes about 90 hours per wheel; a finish cut, 120 hours. Once the finish cut is started, the hobbing cannot be stopped until completed. Accordingly, hobbing machines are equipped with battery power that switches on instantly in case of power failure. The hobbing room and lubricating oil are kept at a constant temperature of 78 to 80 F to limit temperature changes in the wheel during the operation that would seriously affect accuracy of the machining.

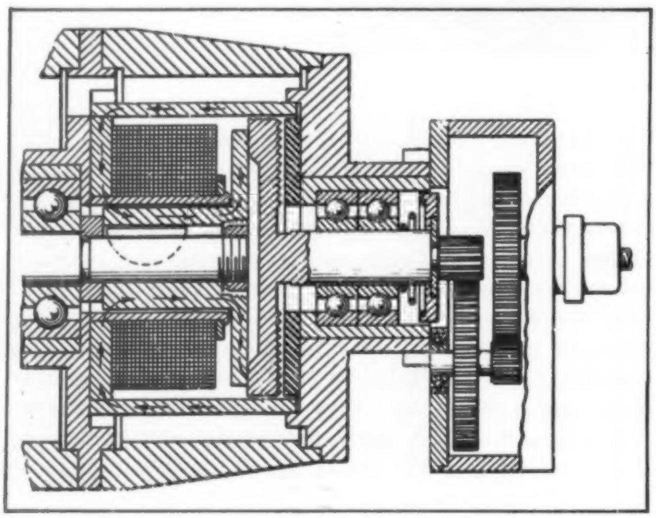
Cyclotron Precision

ACCURACY was the watchword when this 13-foot diameter forging, right, was machined to an accuracy of 0.0015-inch on major dimensions and to a parallel tolerance of 0.0025-inch in the Wheeling, W. Va., shops of Continental Foundry & Machine Company. A pair of these machined forgings are now in service as pole caps in the new 400,000,000-volt Cyclotron near Pittsburgh which will be operated by Carnegie Institute of Technology. The minute steps on the face of the forging measure only 0.017-inch deep. Extremely accurate dimensions and contours are necessary to control the shape of the magnetic field in the cyclotron and to properly direct the paths of the atomic projectiles used in atom smashing.



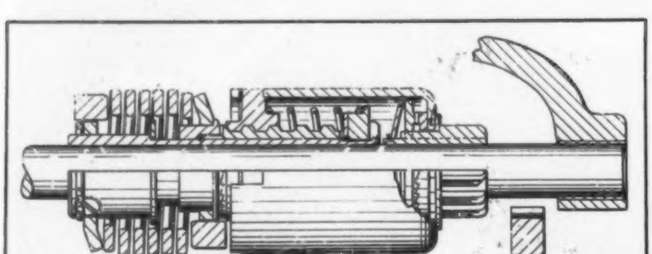
NOTEWORTHY PATENTS

INCREASED BRAKING EFFECT in an electromagnetic clutch and brake mechanism is obtained by grooving the face of the braking member or disk that engages the brake shoe. If the shoe is composed of a relatively deformable material such as



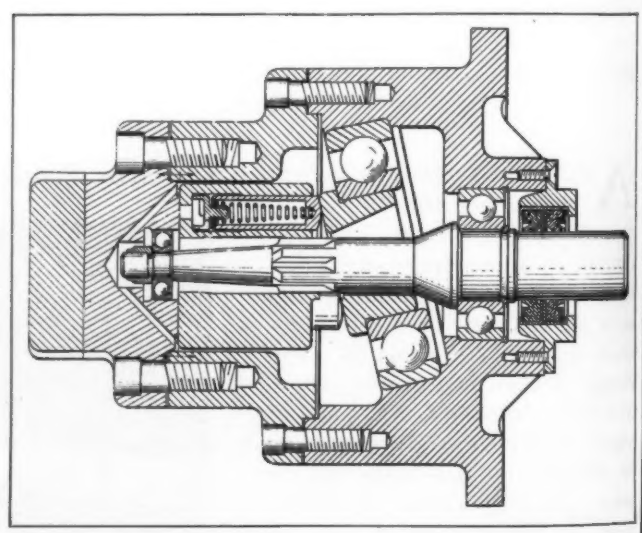
cork, friction surface of the disk is materially increased by the grooves, permitting use of a lighter clutch-disengaging spring which, in turn, allows fewer ampere turns to be wound in the magnetizing winding. Such a device is covered in patent 2,481,028 and assigned to Lear Inc. by William P. Lear.

LIMITATION of the torque transmitted to or from an engine starter, thereby protecting starter and ring gear from accidental overloads, is covered in patent



2,481,324 which Donald L. Miller has assigned to Bendix Aviation Corp. Power from the starting motor is transmitted to a threaded screw shaft through a spring and a saw-tooth clutch. As in a conventional Bendix starter, a control nut attached to a hollow barrel travels along the screw shaft to carry the starter pinion into engagement with the ring gear. If the engine is rotating backward, because of backfiring or some other reason, another clutch between pinion barrel and pinion will slip, protecting the drive from destructive loads. This second clutch consists of two saw-toothed members arranged so that excessive loads cause the spring-loaded clutch faces to wedge apart and permit the teeth to ride over one another.

TILTING OF THE BARREL in swash plate type hydraulic pumps is minimized through the arrangement shown in patent 2,480,069. The rotating pump barrel is splined to the drive shaft at a point such that the resultant of the forces from the pistons and thrust plate is transferred to the drive shaft essentially radially and at approximately the center of the splined section. Such an arrangement also provides

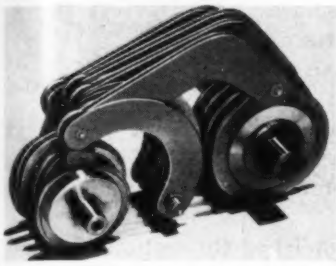


for substantially normal forces between thrust plate and pistons and between barrel and port plate, reducing tendency of the barrel to cock and, consequently, resulting in less wear on the parts. Torsional strain in the compression springs employed to keep the pistons in engagement with the thrust plate is eliminated by mounting the barrel end of these springs in rotating, thimble-shaped bearings which allow the springs to rotate with the pistons. Fred. J. Wright has assigned the patent to Denison Engineering Co.

NEW PARTS AND MATERIALS

For additional information on these new developments see Page 143

Speed Control Mechanism

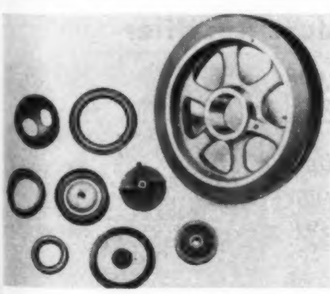


Operating principle of Zero-Max speed control device involves use of crank motion on input shaft to create linear motion which is absorbed on output shaft by series of one-way over-riding clutches. Leverage mechanism is placed between two shaft structures to vary the length of stroke so as to provide infinite speed variations from zero to 4:1 ratio.

Device is compact, can be mounted in any position, has instant response to speed change, which can be made either while operating or stopped, and has high starting torque. Small unit which will handle loads up to 10 in-lb output torque is available with or without case. Manufacturer: Revco Inc., Minneapolis, Minn.

For additional information circle MD 1 on Page 143

Rubber-Metal Bonded Parts



Natural or synthetic rubber and metal can be combined in one bonded unit by Acushnet process to aid in design simplification and cost reduction. Several methods are employed to give desired properties such as resiliency, strength, shock absorption, electrical insulation, corrosion resistance, vibration suppression and protection against abrasion. Service is offered by Acushnet Process Co., New Bedford, Mass.

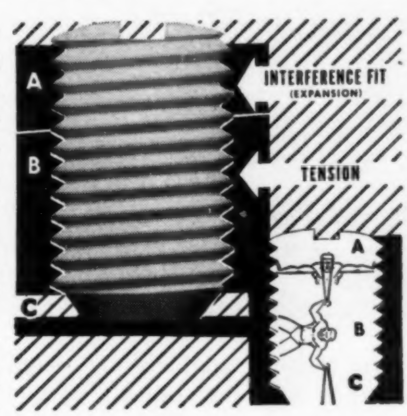
For additional information circle MD 2 on Page 143

Magnesium Extrusion Alloy

Relative insensitivity to notch effect and good impact and fatigue strength are obtained by AK60 magnesium alloy containing 6 per cent zinc and 0.6 per cent zirconium. High strength properties are due mainly to small grain size, insured and maintained by addition of zirconium. Other factors contributing to strength properties are extrusion conditions such as speed, temperature and reduction of cross-sectional area. While used primarily for aircraft floor beams and wings, alloy is applicable to truck and trailer floor sills and parts, textile machinery and materials handling equipment. Manufacturer: Dow Chemical Co., Midland, Mich.

For additional information circle MD 3 on Page 143

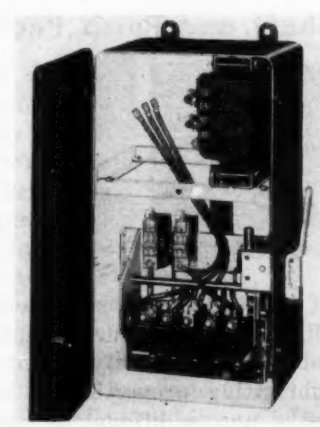
Self-Locking Setscrew



Designed for setscrew applications where vibration is a factor or for regulating and adjustment applications, Zip-Grip setscrew is quickly inserted and tightened. It requires no closer than Class Two fit. Lower part of screw has standard thread, while upper thread section or 'activating area' has larger pitch diameter which creates interference fit or expansion effect against thread flanks. Screws are available in nearly all metals, including hard or soft steel, stainless steel, brass, bronze or aluminum, and are obtainable with any type head. Manufacturer: Set Screw & Mfg. Co., Main St., Bartlett, Ill.

For additional information circle MD 4 on Page 143

Manual Auto-Transformer Starter



Intended for use on squirrel-cage motors, manual autotransformer type starter is available in sizes from 5 hp, 220-440-550 v to 250 hp, 440-550 v for 25, 50 and 60 cycles. Switch mechanism operates with quick-make and quick-break action. Two automatic reset overload relays are standard. Starter is obtainable in form T with air-break contacts and form X with oil-immersed contacts. Air-break type can be converted to oil-immersed opera-

tion merely by adding standard oil tank that is supported beneath switch. Time-delay under-voltage release, which prevents unnecessary opening of switching mechanism on slight voltage dips, is standard on largest starter and can be supplied on other sizes. Furnished on all types, adjustable time-delay mechanism controls length of accelerating period before switching motor from starting to running taps. Manufacturer: Allen-Bradley Co., Milwaukee 4, Wis.

For additional information circle MD 5 on Page 143

Motor Starters and Contactors

Design features of these a-c magnetic starters and contactors include straight-line guided motion of armature and movable contact assembly. Double-break silver-to-silver contacts are readily accessible for inspection and replacement without removing wiring. Coil can be removed with screwdriver.

Sizes 0, 1 and 2 open-type contactors and starters are made to conform to NEMA standards for wiring and mounting dimensions. Holding-circuit interlocks are located to left of vertical centerline. NEMA standardized control and power wiring is furnished with all sizes of contactors and starters. Manufacturer: Square D Co., Industrial Controller Div., 4041 N. Richards St., Milwaukee 12, Wis.

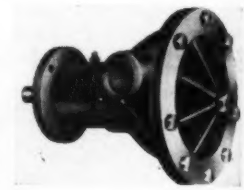
For additional information circle MD 6 on Page 143

Shaft and Pump Packing

Made with alternating rings of solid unoriented Du Pont Teflon and of graphite or mica combined with Teflon, Chemiseal No. 711 shaft and pump packing is particularly adapted to applications where chemical attack from solvents, acids or alkalies or contamination of product involves serious problems. Even under slight gland pressure, graphite-filled rings are forced into leakproof seal against shaft periphery. Natural lubricating properties of both graphite and Teflon make any additional lubricant unnecessary in many applications. Use of solid Teflon rings at each end and alternately throughout packing precludes possibility of flow through packing itself. Teflon is completely inert to all known chemicals at temperatures up to 550 F. Manufacturer: United States Gasket Co., 602 N. Tenth St., Camden, N. J.

For additional information circle MD 7 on Page 143

Positive-Displacement Rotary Pump

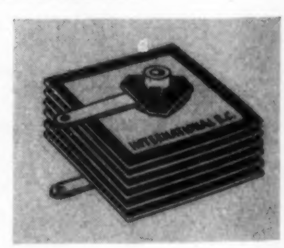


Utilizing floating vanes both radially and laterally, this rotary pump combines turbine action with positive displacement. Light or heavy viscosity liquids or gases can be pumped equally well at continuous operating pressures up to 70 psi or at high vacuum and static head. Unit also can be used as rotary engine or fluid motor with high starting torque and wide speed range.

Radial control is accomplished through use of yoke crosshead and off-center pin; lateral control, by locking vanes inside of rotor. Control mechanism is inside of and revolves with rotor. Vanes do not touch either sides or race as they revolve. Rotor and shaft assembly are carried on preloaded deep-groove sealed ball bearings to prevent end thrust movement. Pressure-controlled mechanical seal is used, and pump operates in either direction with equal efficiency. Available in capacities of 10, 15, and 20 gpm at 0 psi, pump is supplied in cast iron, bronze or of cast iron with bronze fittings. Manufacturer: Chicago Laboratory & Engineering Co., 4140 N. Kedzie Ave., Chicago 18, Ill.

For additional information circle MD 8 on Page 143

High-Voltage Selenium Rectifier



Usable for electronic applications requiring conversion of alternating current to direct current, these high voltage miniature selenium rectifiers require only two soldering connections for installation. They consist of six individually tested and matched cells connected in half-wave circuit and rated at maximum peak inverse voltage of 380 v. Current ratings available are 75, 100, 150, 200, 250, 300 and 350 ma. Manufacturer: International Rectifier Corp., 6809 S. Victoria Ave., Los Angeles 43, Calif.

For additional information circle MD 9 on Page 143

Free-Machining Bronze Alloy

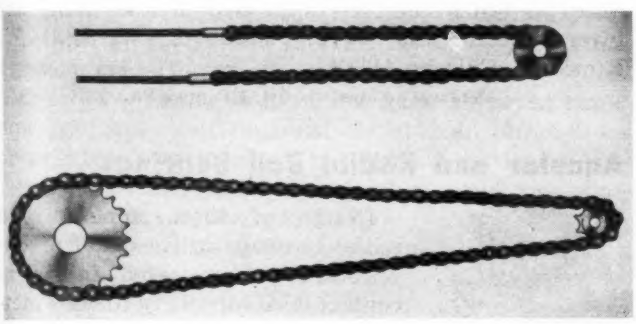
High in lead content, but made without zinc, Mixture "44" bronze alloy has machinabiilty of 90 per cent compared to best free cutting brass rating of 100 per cent or grade B1 phosphor bronze rating of 50 per cent. It is available in rods, sheets, strips and bars. Because of its high lead content, alloy can be

machined to close tolerances at high cutting speeds without burning or scoring cutters. It can be used for screw-machined and similar bronze or phosphor bronze parts and where zinc cannot be tolerated. Manufacturer: Riverside Metal Co., Riverside, N. J.

For additional information circle MD 10 on Page 143

Miniature Mechanical Chain

Specifically designed for use around small pulleys and sprockets, miniature load carrying chain and stud chain are available made from either stainless steel for general usage or beryllium copper for electronic applications requiring nonmagnetic materials. Load-carrying chain of either material weighs approximately ¾-oz per lineal foot, will operate around pulleys as small as 3/16-in. diameter and is capable of passing through 180-degree turn smoothly with-



out binding or slipping. Using stainless steel links with heat treated SAE 4130 pins, load carrying chain has 310-lb yield strength and 428-lb ultimate strength. Beryllium copper chain has 290-lb yield strength and 440-lb ultimate strength.

Stud chain will operate around seven-tooth sprocket with minimum diameter of 0.395-in., either material weighing approximately ½-oz per lineal ft. With stainless steel links, stud chain has yield strength of 100 lb and ultimate strength of 170 lb. With beryllium copper links, it has yield strength of 80 lb and ultimate strength of 140 lb. Manufacturer: Sierra Engineering Co., 123 E. Montecito Ave., Sierra Madre, Calif.

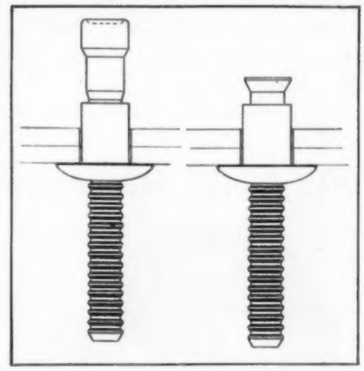
For additional information circle MD 11 on Page 143

Flexible Stainless Steel Hose

Produced from corrosion-resistant No. 20 stainless steel alloy in sizes from 1 to 2⅝-in. ID, this flexible hose is particularly useful on equipment which vibrates or must be moved from time to time. Other applications include fluid-line connections and mechanical parts which must flex. Hose is corrosion resistant to processes which cause formation of sulphuric, formic or acetic acids and other highly corrosive agents. Hose ends can be provided in nonflexible standard tube sizes for use with flanges, or fittings can be welded to ends of hose. Manufacturer: Alloy Tube Div., Carpenter Steel Co., Union, N. J.

For additional information circle MD 12 on Page 143

Pull-Through and Self-Plugging Rivets



Designated as PT pull-through type and 9SP self-plugging type, these blind rivets can be obtained in ⅛, 5/32, 3/16 and ¼-in. diameters. They are furnished regularly in aluminum alloys or cadmium-plated mild steel with either brazier or 100-degree countersunk heads. Pull-through type has no minimum grip limitations, and self-plugging type has grip range of 0.140-in. for any grip increment. Pin tail of rivet has pull grooves to assure positive driving. Positive shank expansion provides ample hole-filling to meet normal requirements. Pin tail is automatically ejected by next rivet and will not become jammed in driving gun. Manufacturer: Huck Mfg. Co., 2480 Bellevue Ave., Detroit 7, Mich.

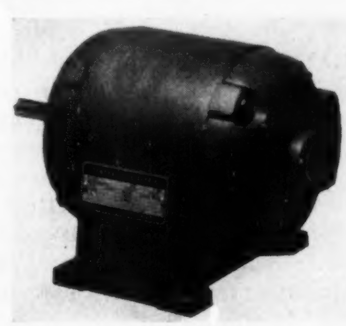
For additional information circle MD 13 on Page 143

Miniature Thermostat

Having electrical capacity of 100 watts at 115 volts, a-c, this miniature thermostat for pilot control applications can be furnished with normally-open or normally-closed electrolytic silver contacts. Extremely sensitive adjustments from 0 to 400 F are made by single adjusting screw. Compact unit weighs only 1/10-oz, and housing measures 1 x 5/16 x ¼-in. overall. It has wire-clinch terminal lugs and is totally enclosed. Manufacturer: George Ulanet Co., 410 Market St., Newark 5, N. J.

For additional information circle MD 14 on Page 143

High-Torque Capacitor Motor



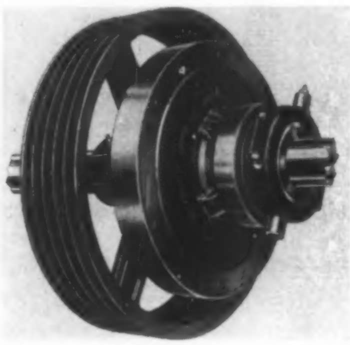
Single-phase Tri-Clad high-torque capacitor motors, having cast-iron construction, are supplied in ratings from ½ to 5 hp. They utilize cast-winding squirrel-cage rotors and are dynamically balanced for smooth operation. To minimize overall dimensions, capacitors are mounted in base of motor and conduit box is replaced by built-in terminal board inside end shield for ease of wiring. Totally-enclosed built-in starting switch protects contacts from foreign matter.

Motors are available in two types which differ only

in starting current, not in output characteristics. Type KCS is a capacitor-start motor designed for operation on 115/230-v, while type KCR is a capacitor-run type for 230 v. Manufacturer: General Electric Co., Schenectady 5, N. Y.

For additional information circle MD 15 on Page 143

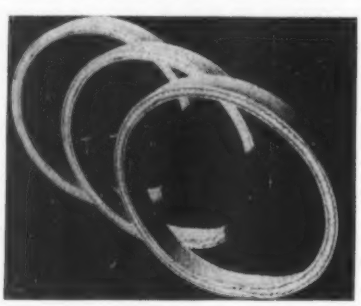
Disk Clutch Mechanism



This type SF disk clutch mechanism consists of standard gear tooth drive designed for starting and stopping flow of mechanical power. The mechanism incorporates an extended bronze-bushed sleeve, tapered on outside diameter to fit tapered bore of Worthington QD sheave. To assemble sheave on sleeve, it is necessary only to remove grease fitting, place sheave over taper and use sheave pull-up bolts to draw it snugly into place. To demount sheave, same bolts are used as jack screws in taps provided for that purpose. Manufacturer: Edgemount Machine Co., Dayton 1, O.

For additional information circle MD 16 on Page 143

Flexible Sleeves and Gaskets

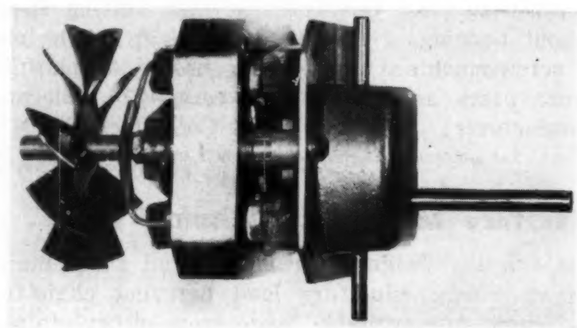


Bonded silicone rubber and Fiberglas by means of mandrel-curing, these flexible sleeves and gaskets are suitable for installations involving high pressures and temperatures. Sleeves are available in sizes ranging from ¼-in. ID to 5½-in. OD with wall thicknesses from 1/16 to 1 in. They can be supplied in lengths up to 3 ft. Having lateral and horizontal strength, sleeves are recommended for applications requiring transmission of hot, cold or corrosive gases and liquids at temperatures ranging from −110 F to more than 400 F. They will not crack, become brittle or deteriorate after long exposure to air, ultraviolet rays or ozone. Manufacturer: Stalwart Rubber Co., 180 Northfield Rd., Bedford, O.

For additional information circle MD 17 on Page 143

High-Torque Gearmotor

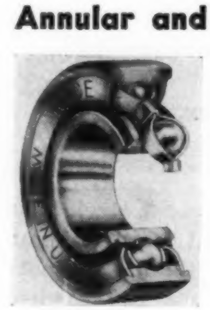
Usable wherever slow speed and high torque are required to drive small electrically operated devices, this gearhead unit measures 3 x 3 x 4½ in. It is



powered by 1/100-hp 1750 rpm four-pole shaded-pole motor with totally enclosed worm gear drive. Standard shaft speeds are 1.95, 3.9, 5.85, 7.8, 11.7 and 17.5 rpm. With right-angle drive, standard speeds are 58, 117, and 175 rpm. Shaft extension is 1 in. Mounted on rear shaft which can also be used as drive is 3-in. diameter fan which directs air over motor. Manufacturer: Electro Engineering Products Co., 4824 W. Kinzie St., Chicago 44, Ill.

For additional information circle MD 18 on Page 143

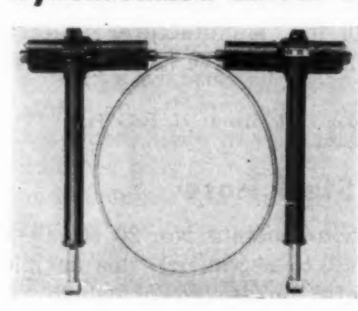
Annular and Radial Ball Bearings



Design of these annular and radial bearings utilizes seal of oil-resistant rubber-coated fabric in contact with especially formed metal shield. Lubricant is effectively retained and foreign materials excluded. Composition seal is used in Nice 1600 and 3000 series bearings. Series 1600 are precision ground annular bearings while illustrated series 3000 are inexpensive unground radials. Manufacturer: Nice Ball Bearing Co., Philadelphia 40, Pa.

For additional information circle MD 19 on Page 143

Synchronized Linear Actuators



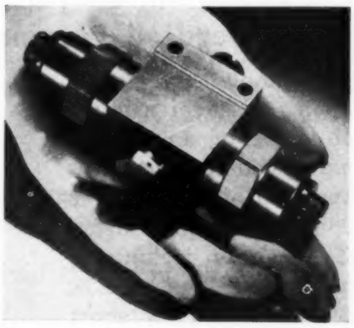
These Lineator electrically operated linear actuators have been redesigned to incorporate mechanical synchronizing feature that permits pairs of units to operate in unison regardless of load variations. They are recommended for use wherever flexibility of electrically controlled device is required to generate thrust. Synchronizing feature provides solution to problem of imparting thrust at identical rate in separate locations, and flexible shaft connecting separate units handles differential power.

Other improvements include nonturn device capable of absorbing full actuator torque in event of striking positive overtravel stop at full voltage with no re-

sisting load, improved radio noise filter and increased motor power. Manufacturer: Airborne Accessories Corp., 25 Montgomery St., Hillside 5, N. J.

For additional information circle MD 20 on Page 143

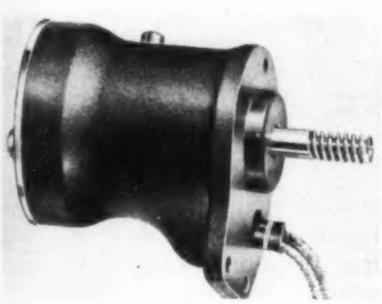
High-Pressure Shuttle Valve



Designed in accordance with drawings AN6277 and AN6278, this 3000-psi shuttle valve meets requirements of specification AN-V-3b and is built in tube sizes 6 and 8. Although produced primarily for aircraft use, valve is applicable to industrial systems where it is desired to supply alternate pressure sources to cylinder or other device as precaution against emergencies, or for other design reasons. Valve has pressure drop of less than 10 psi at rated flow, and will shuttle against closed line. Surge flows or negative pressures will not displace synthetic rubber seat. Manufacturer: Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.

For additional information circle MD 21 on Page 143

Reversible Single-Phase Motor

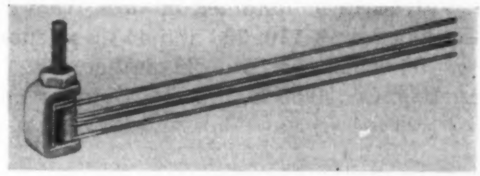


Operating on 115-volt a-c, Model 1201 single-phase, 400-cycle, 1/100-hp motor with permanent capacitor is designed for intermittent duty without maintenance and for reversible applications. It is completely explosion proof, being totally enclosed with copper squirrel-cage rotor and wound field, dipped, baked and wrapped to give protection under extreme conditions. Weighing 8.3 oz, motor is 2-in. high excluding worm gear and has 0.875-in. diameter circular pilot. Unit can be used with valves, lifts, pumps, actuators and like equipment. Manufacturer: Electro-Aire Inc., 11439 Vanowen St., N. Hollywood, Calif.

For additional information circle MD 22 on Page 143

Copper Oxide Rectifier

Maximum stability under extreme operating conditions is feature of compact model CX14 copper oxide rectifier for instrument applications. Gold-to-gold internal circuit arrangement consists of vacuum-

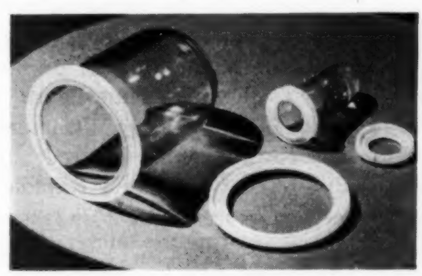


processed rectifier plates with gold contacts, specially-treated gold terminals and copper alloy brackets. Factory-set pressure remains constant under wide temperature variation. Unit is impregnated and sealed to withstand extreme humidity and is fungus-resistant. Measuring ½ x 3/16 x ¼ in., it mounts on one 2-56 x ⅜-in. screw. Manufacturer: Bradley Laboratories Inc., New Haven 10, Conn.

For additional information circle MD 23 on Page 143

Snap-On Gaskets

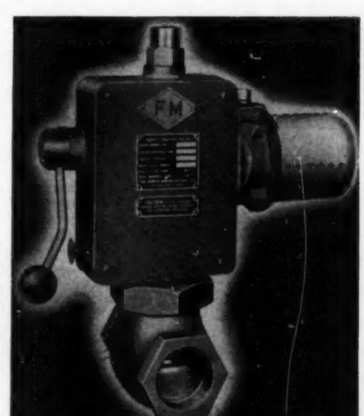
Self-centering, noncorroding and noncontaminating Chemiseal snap-on gaskets are applicable for accommodating deflection and misalignment in assembling sections of conical-end glass pipe and assure smooth



uninterrupted flow of materials inside pipe. They are constructed of solid unoriented Du Pont Teflon material that is impervious to attack from all known chemicals. Manufacturer: United States Gasket Co., 602 N. Tenth St., Camden, N. J.

For additional information circle MD 24 on Page 143

Safety Shutoff Valve



Designed for protection against dangerous accumulations of gaseous or liquid fuels in event of power failure, series 21 Associated Factory Mutual approved safety shutoff valve stops flow of any gas or liquid when current to its solenoid is interrupted. When power is restored, valve remains closed until trouble is corrected, at which time valve is reset manually. Any number of switches wired in series with solenoid can protect installation against explosion hazards resulting from failure of air, gas or oil pressure and general power

NEW PARTS AND MATERIALS

failure. Valve can be obtained in sizes from ¾ to 6 in. and equipped with 110, 220 and 440-v solenoids for 25, 50 or 60-cycle current. Manufacturer: North American Mfg. Co., 4455 E. 71st St., Cleveland 5, O.

For additional information circle MD 25 on Page 143

Cold-Forged Casters

Increased strength and wear resistance are obtained in series 40 casters by cold-forging process. Smooth polished ball race finish is obtained, with contours fitting ball closely to give easy swiveling action. Dust cap prevents dirt from entering raceways and provides means for lubrication retention. Casters can be obtained with nicrosteel, Durastan plastic, molded-on rubber, and cushion or hard rubber wheels in 3, 4, 5, or 6-in. diameters. Load capacities, which vary with types of wheels, range from 175 to 650 lb. Casters can be equipped with thread guards if desired. Manufacturer: Rapids-Standard Co., Grand Rapids 2, Mich.

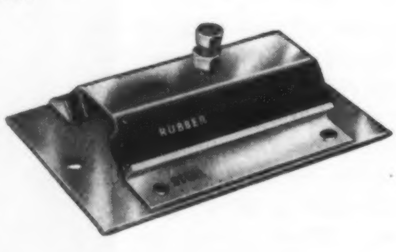
For additional information circle MD 26 on Page 143

Acrylic Molding Powder

Low molding temperatures and fast cycling are possible with Plexiglas VS soft-flow acrylic molding powder for parts which will not be subjected to high service temperatures. In addition to good moldability, it provides water-white clarity. Under standard ASTM test, heat and distortion temperature is 66C. Other physical and chemical properties are similar to those of medium-flow Plexiglas VM powder. Manufacturer: Rohm & Haas Co., Washington Sq., Philadelphia 5, Pa.

For additional information circle MD 27 on Page 143

Vibration Isolator



Drilled, tapped and ready to use, Finnflex SP-9 vibration isolator is easily applicable to all types of machinery and equipment. It consists of steel channel floating in rubber between two steel angles which in turn are welded to steel plate for bolt mounting. Correct angularity between steel and securely bonded rubber enables isolator to approximate flat spot on stress-strain curve of relative slope. Because spring index is not constant, resonance is avoided. Mounts provide for specific load varying from 40 to 3000 lb each and are available in any length up to 30 in. and in four rubber stiffnesses. Manufacturer: Finn & Co., 2850 Eighth Ave., New York 30, N. Y.

For additional information circle MD 28 on Page 143

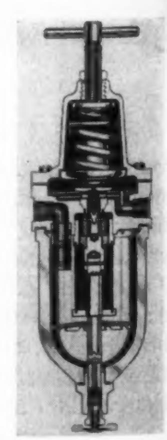
Foot-Operated Pneumatic Valve

Available in ⅜ and ¾-in. sizes for control of single or double-acting cylinders, Mac foot-operated air valve gives operator option of using it as either locking or nonlocking type. Low treadle design and short easy stroke are factors in elimination of operator fatigue in normal nonlocking up and down pedal control. Locking application is obtained by slight forward pressure of toe when pedal is in depressed position. Slight downward pressure of toe will release locking mechanism and permit treadle to return to normal "up" position.

Body design embodies one moving part, a hard chrome spool on which is mounted O-ring seals. These seals enter precision lapped bores of body and end caps to effect instantaneous seal. All tapping ports are on rear to provide convenience for piping. Manufacturer: Mechanical Air Controls Inc., 3049 E. Grand Blvd., Detroit 2, Mich.

For additional information circle MD 29 on Page 143

Filter Regulator



Designed for use with air and any noncorrosive gas at pressure up to 125 psi and ambient temperatures up to 120F, this filter-regulator unit responds quickly to sudden demands for large volume without excessive momentary pressure drop. Directional inlet of permanent filter creates centrifugal action, throwing moisture and entrained solids against bowl, where they run down into quiet zone beneath baffle. Cylindrical screen of reinforced 200-mesh Monel wire removes remaining solids. Surface area is ten times that of pipe opening so that air velocity is reduced and longer cleaning periods are afforded. Large air flow with little pressure drop is assured because of baffle plate and siphon tube action. Unit is 11 inches in height, is easily installed and can be dismantled without removing from line. Manufacturer: C. A. Norgren Co., 222 Santa Fe Drive, Denver 9, Colo.

For additional information circle MD 30 on Page 143

JOHNSON BRONZE

SLEEVE BEARING DATA

Bearing DESIGN

JOHNSON BRONZE

SLEEVE BEARING DATA

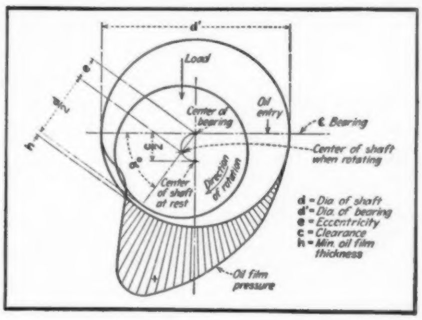
The Lubrication of Sleeve Type Bearings-I

EFFICIENT bearing performance is governed to a great extent by the selection of the bearing material, the bearing design and the method of lubrication. By determining these factors, in relation to the operating conditions, we have the basis for the selection of the proper bearing to deliver the desired performance.

In consideration of the lubrication factor, we must determine the following points: Speed, load, viscosity of lubricant, bearing material, clearance, bearing design, nature of the rubbing surfaces, operating temperature and method of feeding and distribution of lubricant.

While the primary purpose of lubrication is to eliminate mechanical friction and to prevent wear and corrosion, no appreciable amount of power can be transmitted on the best of bearing materials without definite lubrication. Heat develops rapidly when the shaft operates in direct contact with the bearing. Heat is also developed beyond the normal radiating capacity if the viscosity of the lubricant is too high in relation to the speed of the shaft.

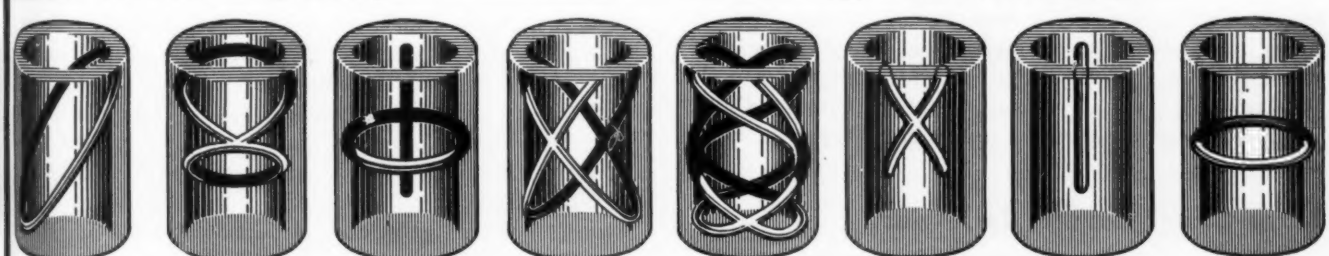
It is a mistaken idea that any type of oil or grease will provide suitable lubrication. The extensive research accomplished by the producers of lubricating materials enables any user of bearings to select the exact type required for his applications. In certain applications it is necessary to provide protection to the lubrication by the use of seals or the design of the bearing. This prevents the lubricant from seeping out with consequent loss and damage to material in process.



With the starting of the shaft the frictional resistance is high. As the speed increases, the lubricant under the influence of the moving part, plus its adhesive qualities, builds up a relatively thick film between the surfaces at the point of nearest approach. The shaft then undergoes a displacement in a direction at right angles to the line of load. The continued speed of the shaft creates a film pressure which increases until an equilibrium is established.

* * *

THE CURVE, as shown at left, represents the distribution of the calculated values of pressure in relation to the bearing reaction. The load-carrying capacity is governed by the maximum pressure that can be developed under the existing conditions. If the load pressure is down—as illustrated—then the curve indicates that the lowest pressure is practically opposite. Obviously this is the proper place for the intake of the lubricant.



A B C D E F G H

Oil Grooving

● The purpose of oil grooves is to assure proper distribution of the lubricant along the length of the bearing in the region just ahead of the loaded area (see above). Complicated and elaborate grooves are in most bearings neither necessary nor useful; they render manufacturing more difficult, and, if not properly placed and executed, may do more harm than good because they may assist in interrupting the oil film.

Deep grooves should always be avoided and both edges of any oil groove should be rounded off to permit free entrance of the oil into the clearance.

Grooves should never be located in the high-pressure area of an oil lubricated sleeve bearing.

In order to obtain quick and uniform distribution of the oil along the length of the bearing, the oil groove should extend axially from the point of entrance and should stop short of the ends of the bearing to minimize oil leakage.

In applications where the direction of rotation changes, spiral oil grooves approaching the high-pressure area, like those shown in (a) and (b), might be expedient.

In many types of split-half bearing, especially when pressure lubrication is used, longitudinal chamfers (relieves) on each side of the bearing halves, are sufficient to secure distribution along the length of the bearing.

As to the points of the oil entry into the oil grooves, they should always be located ahead (relative to the direction of the shaft rotation) of the pressure area; that is, in the unloaded side in bearings with constant load direction.

In applications with changing load direction, the oil entry should be placed at points of minimum oil pressure.

For very long bearings, two oil entrances might be expedient or necessary; if so, they should be connected by one axial distributing groove.

Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

This bearing data sheet is but one of a series. You can get the complete set by writing to—

Johnson Bronze

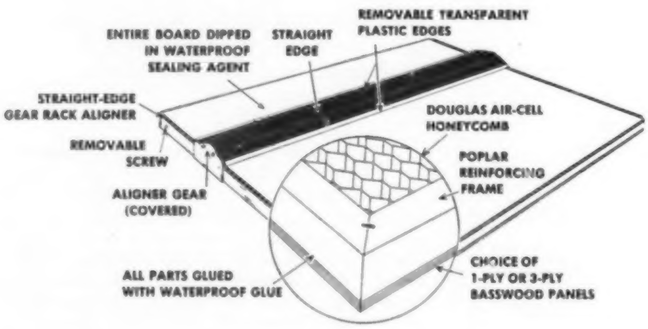
SLEEVE BEARING HEADQUARTERS
525 S. MILL ST. • NEW CASTLE, PENNA.

ENGINEERING DEPARTMENT EQUIPMENT

For additional information on this new equipment see Page 143

Lightweight Drawing Board

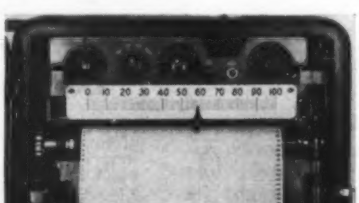
Air-cell type core of this Featherweight panel drawing board reduces weight without decrease in strength. Wood grain of top and bottom plywood panels runs in different directions, and all elements are bonded into rigid unit with waterproof glue and dipped in sealing agent to insure warp resistance. Sizes from 12 x 17 in. to 31 x 42 in. are available. Optional features include straight-edge gear-rack de-



vice which assures horizontal alignment without T-square, easily removable transparent plastic straight edges and washable envelope-type carrying case of water-resistant vinyl-coated fabric. Manufacturer: Cal-Pan Corp., 1111 S. Fremont Ave., Alhambra, Calif.

For additional information circle MD 31 on Page 143

Strip-Chart Strain Recorder

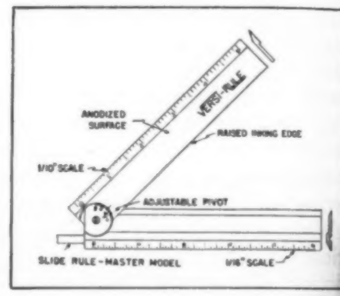


Features of this strip-chart strain recorder for use in stress analysis with bonded resistance wire strain gages are: Easy readability with 9½-in. wide chart scale, two chart speeds of 6 and 180 in. per hour and accommodations for two-arm and four-arm strain bridge. Slowly varying strains can be recorded for as long as ten days without changing chart. Instrument is special adaptation of Leeds & Northrup Speedomax type G, model S recorder with simple adjustments for strain gage characteristics, strain ranges and Wheatstone bridge circuit.

Microinch ranges of recorder are 1000, 2000, 5000 and 10,000 on 6-v bridge and 2000, 4000, 10,000 and 20,000 on 3-v bridge. Continuous gage factor adjustment covers range from 1.7 to 2.2. Response to full scale unbalance is approximately 3 seconds. Manufacturer: Baldwin Locomotive Works, Philadelphia 42, Pa.

For additional information circle MD 32 on Page 143

Aluminum Rule-Gage-Calculator



Slide rule, depth gage, beveled rule, protractor, triangle and square are incorporated in one compact unit in Master Model Versi - Rule. Features include machine - graduated fractional and decimal scales with beveled ruling edges, adjustable pivot friction joint and raised inking edge. It will operate with T-square. Beveled edges lie flat for all positions of rule. Constructed of lightweight aluminum alloy with alumilite finish, rule measures 5⅜ x 1¼ x ⅛-in. when closed and is packed in leather pocket case. Three models are available: Above-mentioned Master Model, Senior model which includes all above features except slide rule and Standard model which incorporates all tools but depth gage and slide rule. Manufacturer: Dall Inc., 720 Washington Ave. S.E., Minneapolis 14, Minn.

For additional information circle MD 33 on Page 143

Portable Electric Tachometer

Assuring accuracy within 0.5 per cent of full scale value in measuring either surface or shaft speeds, M1200-F portable electric tachometer and cutmeter has range of 0 to 25,000 rpm, or surface speeds from 0 to 5000 fpm for continuous or intermittent use. Since generating and indicating instruments are separate, steady readings can be obtained even under adverse conditions. Unit makes possible continuous indication so that speed changes or variations can be observed readily. In many applications, indicator can be permanently mounted in convenient remote location. Standard cable connecting generator and indicator can be lengthened to as much as 500 ft.

Tachometer is completely self-contained and requires no batteries or other external power source. It consists of d-c magneto generator that produces voltage directly proportional to speed being measured,

calibrated in conjunction with easily-read laboratory type indicator having four overlapping speed ranges. Two styles of driving points, extension shaft and cutmeter wheel assure easy application of instrument to practically any type of rotating equipment. Entire assembly is housed in quartered oak carrying case. Manufacturer: Electric Tachometer Corp., 2218 Vine St., Philadelphia 3, Pa.

For additional information circle MD 34 on Page 143

Reproduction Machine

Capable of producing up to 10,000 sq ft of prints during ordinary working day, Model 50 Whiteprinter machine reproduces anything drawn, typed, printed or written. Prints are delivered flat and dry, neatly stacked in convenient receiving tray on front of machine.

Prints ranging from postcard size to 42 in. wide by any length can be produced on sensitized paper, film or cloth furnished in form of cut sheets or roll stock. In addition to usual black lines on white background, machine can produce prints in 20 combinations of colored lines and tinted paper to make possible color keying. Machine requires no plumbing connections or ventilation ducts and is odorless in operation. Manufacturer: Charles Bruning Co., 4754 W. Montrose Ave., Chicago 41, Ill.

For additional information circle MD 35 on Page 143

Electronic Standard Cell

Available for any specified output from 0 to 100-volts d-c and for any load up to 30 ma, this standard electronic cell is usable for instrumentation requiring accurate and highly stable voltage. Output voltage is constant to better than 0.1 per cent and with ripple less than 0.01 per cent, throughout input range of 75 to 135 volts a-c at frequencies from 50 to 400 cps. Cell can be used over wide temperature range, is not subject to freezing and is not damaged by momentary short circuits. It can be used either as reference voltage in bridge or potentiometer circuits or for supplying current continuously as instrument power supply. Cells designed on same circuit principles for output voltages above 100 volts, d-c, for operation on other input voltages, higher current drains, or with nonstandard chassis construction are available on special order. Manufacturer: Hastings Instrument Co., Box 1275, Hampton, Va.

For additional information circle MD 36 on Page 143

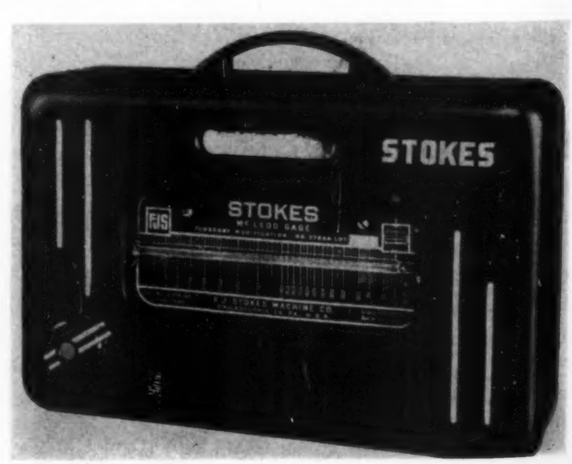
Diazotype Paper and Cloth

Plastic surfaced intermediate Tecnilith papers and Tecnicloth cloths are available in both sepia-line and black-line formulations for use on whitprint machines. Printed in reverse from original tracing, Tecnilith prints are said to produce better blue or black-line prints than those produced from original tracing at equally fast speeds. Image will not bleed, offset on other prints or become brittle or yellow with age. Plastic surface protects paper from dirt; but in event of soilage, it can be cleaned with soap eraser or moistened cloth. Print closely resembles photgraphic reproduction and can be corrected easily if desired. Having highly transparent finely woven tracing cloth base, Tecnicloth is durable. Manufacturer: Tecnifax Corp., Holyoke, Mass.

For additional information circle MD 37 on Page 143

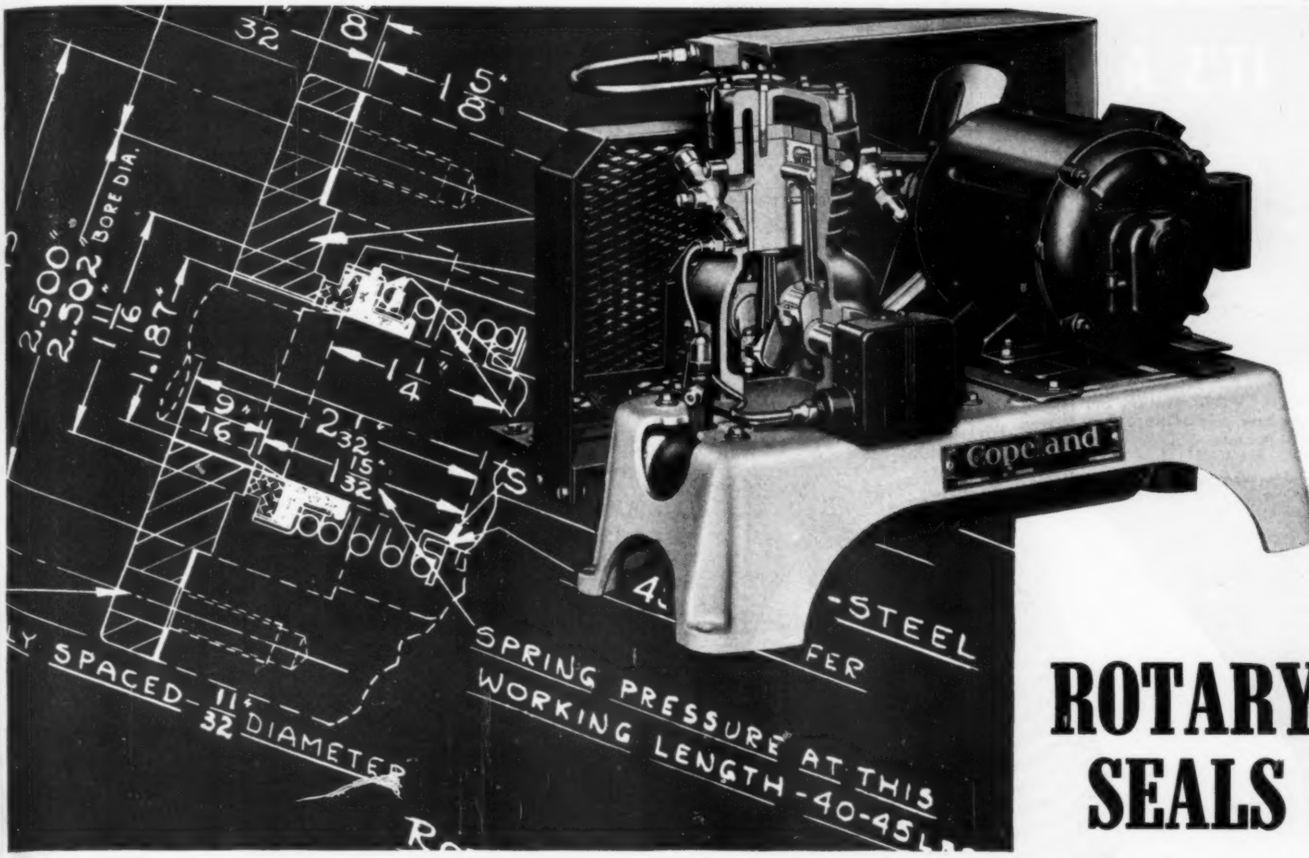
High-Vacuum Gage

Calibrated in millimeters, with lowest scale division of 10 microns, this McLeod type high-vacuum gage has working range of 0.05 to 50 mm. It performs well in applications subject to contamination



during operation. Portable unit weighs $6\frac{1}{2}$ lb, with overall dimensions of $7\frac{1}{2}$ x $11\frac{7}{8}$ x $4\frac{1}{2}$ in. Gage is housed in plastic case and is furnished complete with 1 lb of mercury. Manufacturer: F. J. Stokes Machine Co., 5900 Tabor Rd., Philadelphia 20, Pa.

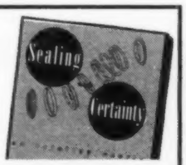
For additional information circle MD 38 on Page 143



ROTARY SEALS

give Copeland Condensing Units *Certainty* in *Sealing!*

THE ROTARY SEAL PRINCIPLE



is the original approach to a practical solution of a universally troublesome problem. Our booklet "SEALING WITH CERTAINTY" explains and illustrates the principle. We're glad to send it to you without obligation.

Copeland's slogan is "Dependable Electric Refrigeration"—and they back it up with painstaking attention to every part that goes into the many models in their extensive line. They can't take chances on Seal failure—and with Rotary Seals they have the assurance of expert engineering, careful manufacture, and a record of proven sealing performance in many fields. Those factors add up to *Certainty in Sealing* for trouble-free, day-in-and-day-out satisfaction in the use of equipment, even under the hardest conditions.

The particular Seals used in Copeland compressors are special adaptations of the basic Rotary Seal principle, developed to fit the particular requirements. Our engineers can help to provide the right answer for *your* sealing problem, whatever your field. For best results and greatest economy, call us in on the job early. The variety of our experience with successful applications of Rotary Seals often indicates the simplest design approach.

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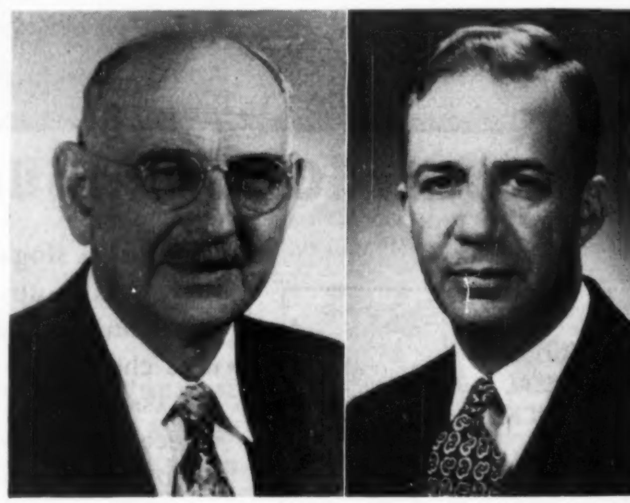
MEN OF MACHINES

Appointment of **George F. Noltein** as director of research of its Superior Engine Division, Springfield, Ohio has been announced by The National Supply Co., Pittsburgh. At the same time announcement was made of the appointment of **Harvey W. Hanners** to succeed Mr. Noltein as chief engineer. Mr. Noltein, who has a long and successful record as an engineer and designer in the diesel engine field both in Europe and the United States, joined National Supply's Superior Engine Division in 1927 and has



George F. Noltein   Harvey W. Hanners

been chief engineer since 1937. He received his engineering education in Germany and Russia, graduating from the Institute of Technology, Danzig, and the Moscow Institute of Technology. Mr. Hanners, who has been associated with Superior Engine Division since 1941 and assistant to the chief engineer for the past two years, graduated from the University of Wisconsin and has specialized in the manufacture and development of both diesel and gas engines.

•

Announcement has been made of the appointment of **Waldemar Naujoks** as special projects engineer of Ladish Co., Cudahy, Wis.

•

F. H. Boor has joined the engineering staff of the Wisconsin Axle Division of the Timken-Detroit Axle Co., Detroit. Formerly in charge of the engineering department of the Fairchild Mfg. Co., Mr. Boor has

How to put precision right on the spindle nose

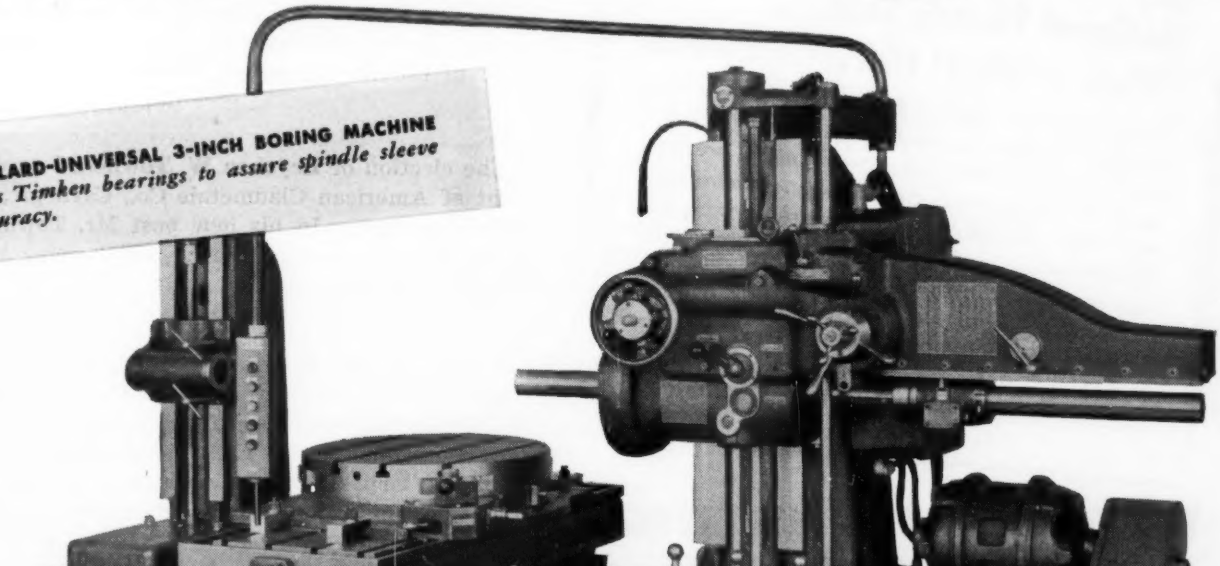
SPINDLE precision is assured in the Bullard-Universal Horizontal Boring Machine by using Timken® tapered roller bearings on the spindle. And Timken bearings in the drive mechanism keep shafts turning and gears meshing smoothly under all operating conditions.

Timken bearings hold spindles in positive alignment under the heaviest loads. Bearings are accurately preloaded to predetermined engineering limits. The tapered construction of Timken bearings carries radial and thrust loads in any combination, eliminating end-play.

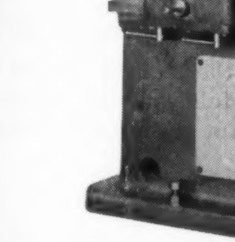
Line contact between the rolls and races of Timken bearings provides maximum load carrying capacity. The rolls and races are made from Timken fine alloy steel for toughness—then case-hardened to provide wear-resistant surfaces. Because Timken bearings are manufactured to extreme precision and finished to almost incredible smoothness, friction virtually is eliminated.

Remember, no other bearing can bring you *all* the advantages you get with Timken tapered roller bearings. Whether you build machine tools or buy them, always be sure they're Timken bearing equipped. Look for the trade-mark "Timken" on the bearings. The Timken Roller Bearing Company, Canton 6, Ohio. Cable address: "TIMROSCO".

This symbol on a product means its bearings are the best.



BULLARD-UNIVERSAL 3-INCH BORING MACHINE *uses Timken bearings to assure spindle sleeve accuracy.*



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Finishing to incredible smoothness accounts for much of the precise, smooth rolling performance of Timken bearings. This honing operation is typical of the amazingly accurate manufacturing methods at Timken.

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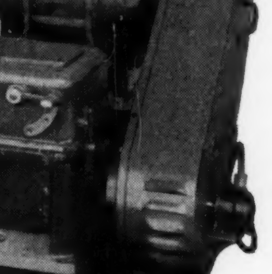
TAPERED ROLLER BEARINGS



NOT JUST A BALL — NOT JUST A ROLLER — THE TIMKEN TAPERED ROLLER — BEARING TAKES RADIAL AND THRUST — LOADS OR ANY COMBINATION

had extensive experience in the design of gears and differentials used in farm equipment, trucks and power shovels.

•

E. J. de Ridder recently joined the staff of technical service engineers at Reynolds Metals Co., Louisville, Ky. He was connected with I. G. Farben Lightmetal Division from 1926 to 1945.

•

J. Packard Laird is now research engineer with E. I. Du Pont de Nemours & Co. Engineering Research Laboratory Experimental Station, Wilmington, Del. Mr. Laird, who was formerly associated with Buehler & Co., has been a contributor to MACHINE DESIGN, his latest article appearing in the April, 1949 issue.

•

Marvin W. Smith, president of the Baldwin Locomotive Works, Philadelphia, has been elected a director of the Westinghouse Electric Corp., Pittsburgh.

•

The election of **Leyshon W. Townsend** as vice president of American Cladmetals Co., Carnegie, Pa., has been announced. In his new post Mr. Townsend will be in charge of manufacturing, engineering and product development.

•

Ludger E. La Brie has been appointed chief engineer of the Warner Aircraft Corp., Detroit. He had been chief engineer of the Lockheed Division of the Hydraulic Brake Co., a subsidiary of Bendix Aviation Corp. In his new position Mr. La Brie will supervise the engineering and development of aircraft hydraulic controls.

•

Roger S. Warner has joined the staff of Arthur D. Little Inc., Cambridge, Mass., following his resignation as director of engineering for the Atomic Energy Commission.

•

Wu Wai Chao has been appointed to the staff of the Polytechnic Institute of Brooklyn to expand its graduate offerings in the newest developments in thermodynamics and the mechanics of fluids. A graduate of Tung-Chi National University of China, Dr. Chao spent the last four years at the Massachusetts Institute of Technology where he earned his master's and doctor's degrees.

•

Recently appointed to head the hydraulic department of Holyoke Machine Co., Holyoke, Mass. is **L. H. Perry,** who was formerly with James Leffel & Co. as mechanical engineer designing hydraulic turbines for both government and private power plants. Prior to that, Mr. Perry was with Charles T. Main, Engineers, of Boston, supervising the de-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Bulletin 49-55

sign of hydraulic and mechanical equipment for government power plants. He was also for a number of years hydraulic engineer in charge of the water wheel department of Rodney Hunt Machine Co.

•

Rudolph A. Goepfrich has been appointed chief engineer of the automotive brake department, Bendix Products Div., Bendix Aviation Corp., South Bend, Ind. **Clark R. Lupton** succeeds Mr. Goepfrich as assistant chief engineer.

•

Several changes have been announced in the executive staff of the B. F. Goodrich Co. engineering division. **H. E. Cook,** chief engineer for many years, has been assigned to major engineering problems including direction of design and construction. **Frank Brown** has been appointed plant engineer of all Akron plants, **George Murphy** to the same post at Plant 4, and **Clifford R. Augden** has been named superintendent of shops.

•

Appointment of **F. B. Law** as manager of manufacturing of the Aeronautic and Ordnance Systems Division of General Electric Co. has been announced. A mechanical engineering graduate of George School of Technology Mr. Law, who has been with the company since 1926, succeeds **Frank T. Lewis.**

Investment Casting

(*Concluded from Page* 99)

and it is not always possible to maintain long, thin sections perfectly straight. Large areas are not always possible to cast free from imperfections. Areas which must be free from defects should be so marked. Cost of parts is affected directly by the degree to which minor imperfections are permissible. Surface roughness generally can be held to 70 to 80 microinches, rms, without added expense. With plastic patterns surfaces with roughness from 10 to 20 microinches, rms, have been produced.

Angular accuracy is often difficult to hold but, as a rule, a tolerance on angles of plus or minus ½-degree can be held. It is practical, however, on some parts to hold this to plus or minus ¼-degree, *Fig.* 16. With precision steel pattern molds, gears with plus or minus 0.001-inch on tooth dimensions and plus or minus 10 minutes angular tolerance have been made.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:

Allis-Chalmers Mfg. Co. (*Figs.* 7, 8) ..... Milwaukee, Wis.
Arwood Precision Casting Corp. (*Figs.* 9, 10) .... Brooklyn, N. Y.
Haynes Stellite Co. (*Figs.* 6, 13, 14, 15, and 16) .... New York, N. Y.
Illinois Precise Casting Co. (*Fig.* 2) .... Chicago, Ill.
International Nickel Co. Inc. (*Figs.* 3, 4, 5, 11, and 12) .... New York, N. Y.
Jelrus Co. Inc., The (*Fig.* 1) .... New York, N. Y.
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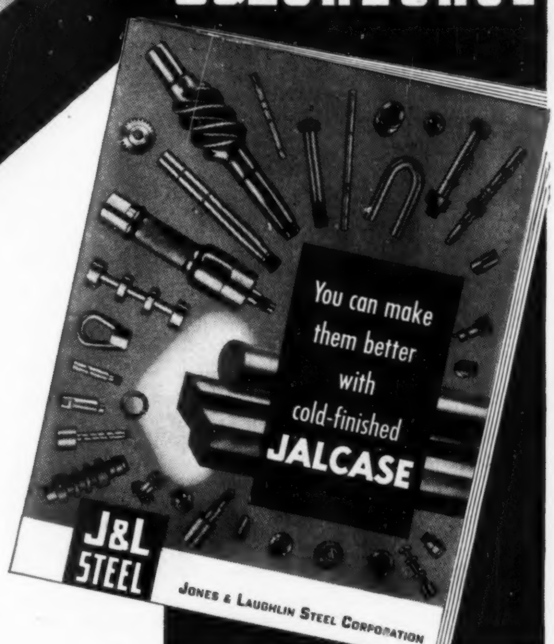

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NEWS OF MANUFACTURERS

The **Porter-Cable Machine Co.,** Syracuse, N. Y., manufacturer of Speedmatic and Guild portable electric tools, has purchased the manufacturing rights and facilities of the **Sterling Electric Tool Products Co.,** Chicago, Ill., for the production of the Sterling portable electric and pneumatic sanders. The inventory, tools and production facilities which have been purchased will be moved to Syracuse, where the manufacture of Sterling products will be resumed.

♦

Purchase of all outstanding stock of **Electrical Reactance Co.,** manufacturer of radio capacitors and resistors, by **Aerovox,** New Bedford, Mass., manufacturer of electrical and radio-electronic test instruments, etc., has been announced. Electrical Reactance plants at Franklinville, N. Y., Jessup, Pa., and Myrtle Beach, S. C., will be operated as a wholly-owned subsidiary.

♦

Miehle Printing Press and Mfg. Co., Chicago, Ill., manufacturer of printing presses, has purchased the capital stock of **Star Electric Motor Co.,** Bloomfield, N. J., manufacturer of electric motors and generators. The present management will undergo no changes for the present and will assume, in addition, the management of the **Kimble Electrical Division** of Miehle located at Burlington, Iowa.

♦

Whitney Chain & Manufacturing Co., Division of **Whitney Hanson Industries Inc.,** has moved to new and larger headquarters at 3317-25 W. Newport Ave., Chicago, Ill. The Whitney division manufactures a complete range of roller, silent and conveyor chains, couplings and sprockets.

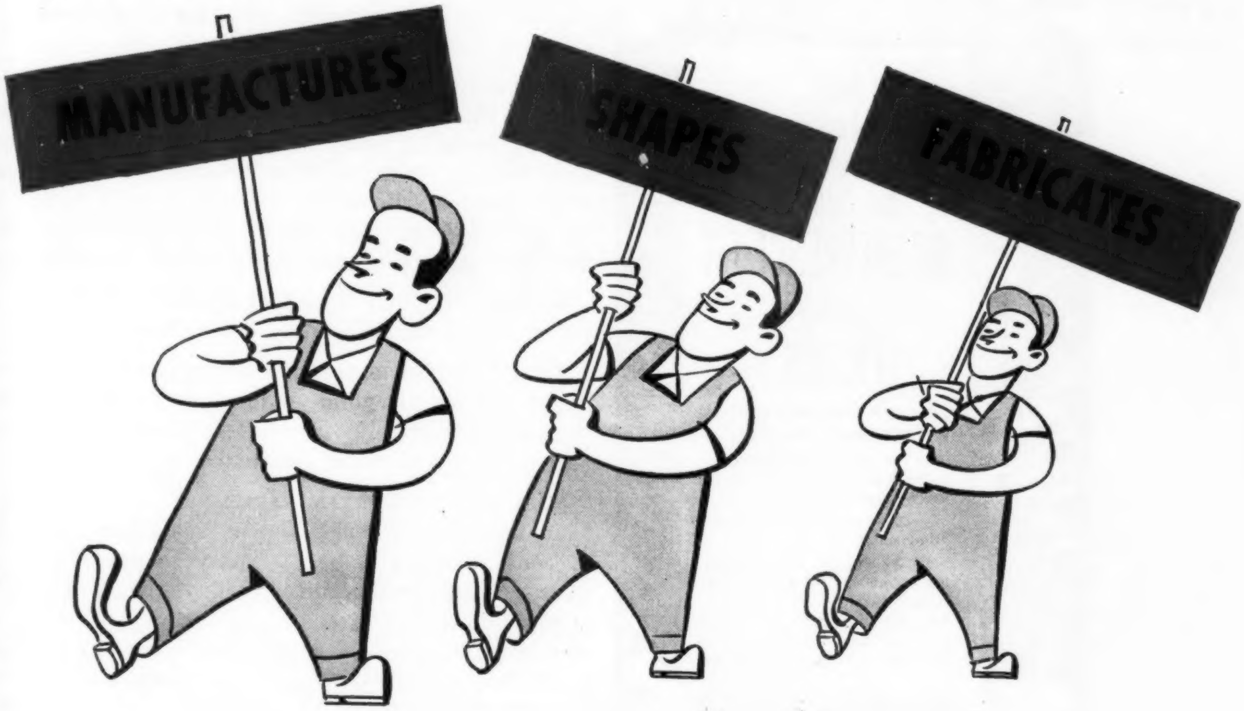
♦

Pacific Airmotive Corp., Burbank, Calif., and **Stratos Corp.,** Farmingdale, Long Island, a wholly-owned subsidiary of the **Fairchild Engine and Airplane Corp.,** have announced an agreement whereby aircraft manufacturers will be able to obtain a fully coordinated pressurization and air conditioning installation for military and commercial aircraft. The equipment is adaptable to both propeller and jet aircraft.

♦

The **Cleveland Graphite Bronze Co.** has formed a Canadian subsidiary, **Clevite Ltd.** which has purchased a factory at St. Thomas, Ontario, and will begin production of sleeve type automotive bearings early in 1950.

Ostuco does all three...



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Hydraulic Oil

(*Continued from Page* 86)

ated clay types are not required because the oxidation inhibitor prevents formation of acidic materials requiring removal. On the other hand, clay filters remove the inhibitors and thus have detrimental instead of beneficial effects on such oils.

Breathers or Vents: Reservoirs are frequently open to the atmosphere, even in unusually dirty locations, permitting entry of dust or dirt. In addition to accelerating oxidation this contributes to excessive wear, particularly where the dirt is of a gritty and abrasive nature. Air filters should be provided to minimize such potentially destructive contamination.

Covers: Tightly fitted, gasketed covers are obviously essential to keeping the system clean.

Drainage Provisions: Reservoir drain plugs should be at the lowest point, making it possible to drain the tank completely and obviating pumping, dipping and mopping in order to remove all oil.

Soluble-Oil Contamination: This continues to be a serious problem in many machines. This type of contamination is usually easy to detect since the hydraulic oil first becomes cloudy, then milky in appearance. However, soluble-oil contamination is not always so obvious. A typical instance, which follows, is illustrative of the persistence sometimes required to trace the source of hydraulic-oil contamination.

Investigate Design Before Condemning Oil

Samples of oil from certain types of well-designed machines showed much more rapid increase in neutralization number than those from other machines in the plant. However, there was no cloudiness or water content to indicate the presence of soluble oil. A check of the hydraulic-oil reservoir indicated that there was no possibility of contamination by drip onto the cover.

Further investigation revealed that the hydraulic-pump drive chain dragged along the cover of the soluble-oil sump, and when the soluble oil overflowed onto this cover the emulsion was carried by the chain and splashed against the pump shaft and, apparently, the centrifuging and impingement forces separated out the oil from the water. The particular type of soluble oil used had unusual wetting and penetrating properties. It was deduced that it crept along the pump drive shaft and so entered the hydraulic pump chamber. Steps were taken to prevent flooding of the soluble-oil sump cover and also to prevent the chain from dragging. After these measures were taken the hydraulic oil showed no further abnormal increase in neutralization number.

This instance is covered in detail for two reasons. First, because it indicates the amount of thought required on the part of designers to prevent contamination of hydraulic oils; second, because it demonstrates the fallacy of thoughtless condemnation of oils on the basis of isolated cases of trouble with-

NATIONAL OIL SEAL LOGBOOK

DIRT EXCLUDERS FOR HYDRAULIC & PNEUMATIC ACTUATED DEVICES

Truck hoists and hydraulic lifts of all kinds usually must perform under extreme operating and surrounding conditions. Piston shafts sometimes become caked with heavy abrasives; fine dust, muck and dirt are common. Shafts are many times subject to angular thrusts which cause considerable movement off center (Fig. 1). These conditions can result in the entry of copious quantities of extraneous matter into the cylinder causing excessive wear or failure of pressure seals whether of "O" ring or packing gland type.

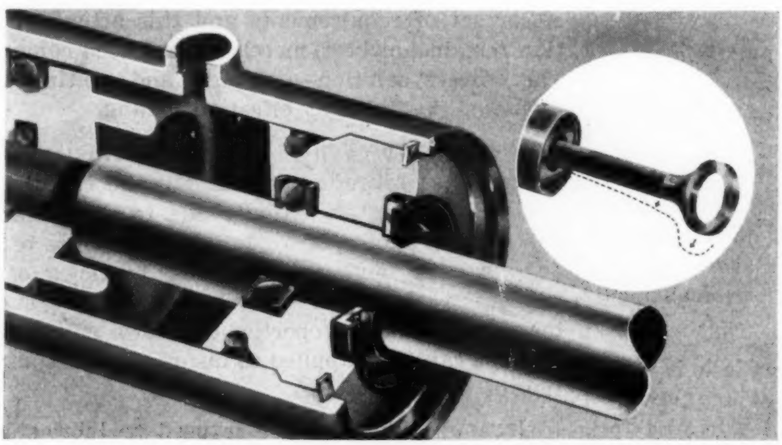

Fig. 1. Typical section of hydraulic actuated mechanism showing use of National Seal as dust excluder—note "O" rings and back-up rings.

Rugged, yet flexible, Syntech* oil seals of the type illustrated (Fig. 2) provide excellent protection for such equipment. These seals act as a wiper on the reciprocating shaft to clear away extraneous abrasive matter. Where ice or heavy caking of dirt is encountered a combination of bronze ring scraper and a Syntech* seal (Fig. 3) can be employed—the bronze scraper to break up and remove the heavy cake and the oil seal to clean the shaft.

These National Syntech* Seals have been extremely successful in applications of this kind where leather type seals and other devices have failed. The Syntech* rubber is flexible enough to follow the shaft eccentricities, yet sturdy enough to keep from buckling under as the shaft moves back and forth, even under extreme shaft eccentricities.

National Syntech* Seals in combination with "O" rings provide a close approach to the ideal for hydraulic or pneumatic actuators of all types and sizes. However, leather or other sealing member materials are supplied in a few special applications. National can also supply "O" rings and back up rings as shown (Fig. 1) to meet the most rigid specifications in a complete range of sizes. Your inquiry will receive prompt attention.

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Fig. 2. National 340,000 Series springless type seal. A good design for use in hydraulic or pneumatic actuators.

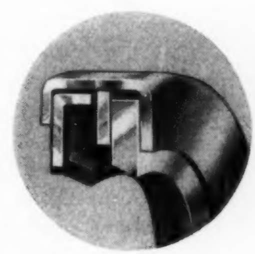

Fig. 3. Combination National Syntech* Seal with bronze scraper ring.

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out thorough investigation to determine the true cause of apparent oil failure.

NONSOLUBLE CUTTING-OIL CONTAMINATION: Contamination by nonsoluble cutting oils is another common problem. It is particularly bad because corrosive components of such oils may cause severe corrosion of system parts, while fatty components seriously impair resistance to oxidation.

Effective Sealing Imperative

Intermingling of cutting oils with lubricating and hydraulic oils is most difficult to control on multiple-spindle automatics. It is sometimes desirable to use dual- or triple-purpose oils in such cases. However, such oils must, of necessity, involve a compromise between chemical stability, demulsibility and nonfoaming characteristics on the one hand and tool protection and satisfactory work finish on the other. Something must be sacrificed as to ability to meet either set of requirements and this attempt at solution remains, at best, merely an enforced compromise. The correct solution—complete and continued separation of the fluids by effective sealing.

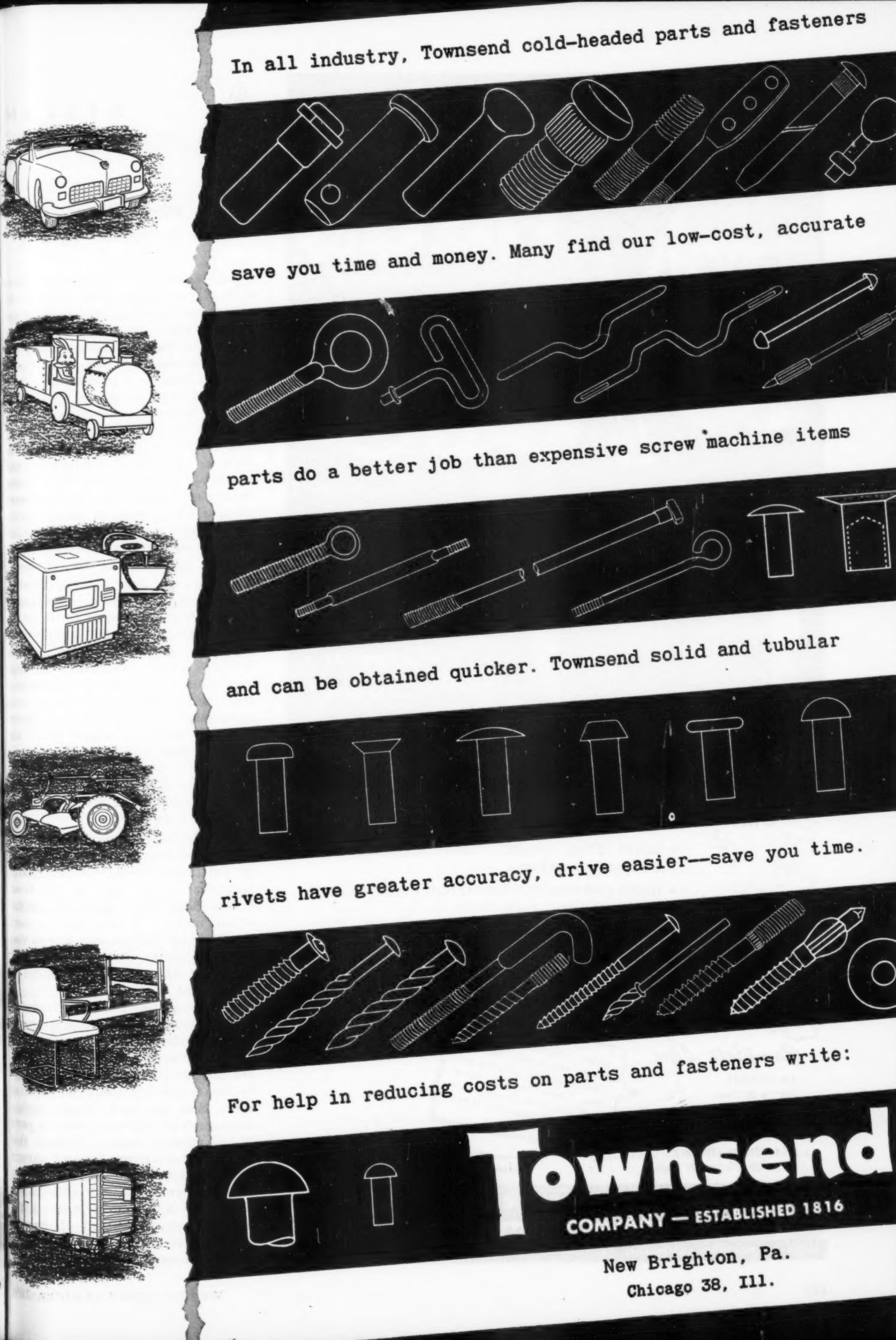
HOSE, PAINT, AND GASKETING MATERIALS: Some deposits in hydraulic systems that are charged to oil oxidation are often actually due to deterioration of hoses, or to improper kinds or careless use of paints or gasketing materials, causing particles of these substances to be picked up by the oil. All such materials must have properties of known suitability and must be carefully applied to insure against contamination of the oil.

GREASE: Where grease is used to lubricate pump bearings, it sometimes leaks into the pumps and contaminates the hydraulic oil. To prevent this the bearings must be properly sealed and grease of proper consistency and stability must be used. The frequency of application and amount of grease used should be carefully determined to fit the operating conditions.

AIR IN THE SYSTEM: The various means by which air is introduced into hydraulic systems and the adverse effects of air have been mentioned. To prevent introduction of air effective shaft and rod seals must be provided and properly maintained, oil return lines should be submerged, connections in suction lines must be airtight, and the pump suction should be close to the bottom of the reservoir to insure as complete submersion as possible. Automatic bleeders should be provided to facilitate elimination of air that enters the system in spite of precautions.

VIBRATION: Air leaks into hydraulic systems and oil leaks from them frequently develop due to loosening of line connections because of vibration. Where this is severe, flexible connections are required. On some large, high-pressure systems, welding has been used to ensure freedom from leakage due to loosening of connections by vibration.

The foregoing observations and suggestions may be of some help to designers and builders in making hydraulic systems more efficient, reliable and economical.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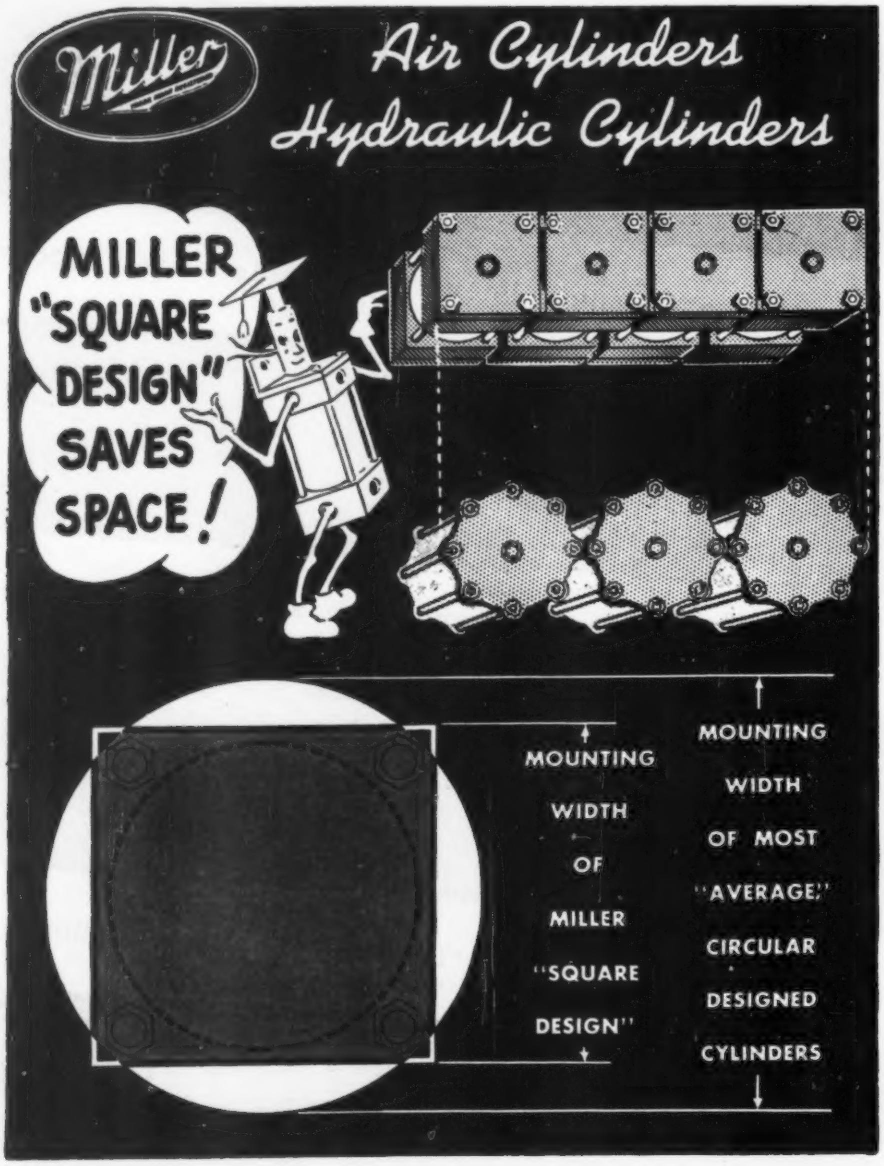

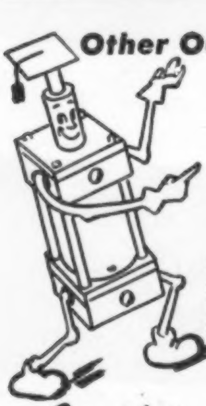

DESIGN ABSTRACTS

What's Wrong With Automobile Styling?

ALL of us will agree that the lines of an automobile should be smooth and flowing, but I believe designers have over-emphasized this smooth, flowing principle to a point where the cars of today are actually soft and mushy. I predict that there will be a trend in the future toward designs that are more crisp and well defined. We have found this to be true in the design of other products, such as household appliances and business machinery. We have found that the buying public welcomes these crisper, slightly more severe shapes, as a relief from the so-called streamlined shapes that have been so much in vogue for the last ten years.

Another fact that warrants discussion is the practice of over-emphasizing functional features such as fog lights, tail lights, and license-plate brackets. Making each of these a separate center of interest of high importance gives one the impression that the car was designed by several independent groups of designers, working entirely apart and not knowing what the other groups were up to and not knowing what the general style and lines of the car were to be. As a matter of fact this might well have been the case.

They certainly have simplified the problem of louvers today. They merely put little, round, chrome-plated portholes on the side of the hood. I have seen these on so many different makes of cars that I can't recall at the moment who it was that first used them. But I will say that they look as good on one model as they do on another.

Chrome plating is most certainly a necessity, and if its use is restricted to the parts of the automobile for which it was originally developed, it provides a very interesting accent. To be specific, I think we can all agree that its hard, tough surface belongs on the bumpers and on parts of the radiator grille. I think that the use of a narrow bead of chrome around the windows provides a note of refinement, although this is not necessarily a functional use. I like the use of the single horizontal chrome strip that is used on the sides of most cars today. Besides

being a nice accent that makes the cars look longer and lower, these strips perform a very definite function. Serving as efficient rub rails, they protect the body against scratching. However, the effect of these horizontal strips is usually spoiled by the use of the large chrome-plated, wing-shaped pads that are placed directly below them on both the front and rear fenders. I suppose it can be argued that the pad on the rear fender is needed as a stone shield; however, I have noticed that some models get along without tnis added protection and seem none the worse for wear because of the lack of this shield and actually look better for it.

I don't like the use of long chrome-plated fins on the tail lights. I don't think that a wide bank of chrome ribs running the length of the hood or down the back of the rear deck enhances the appearance of a car. I don't like the heavy, superfluous chrome escutcheon around the license plate and stop lights in the back of the car. I think there is a deplorable tendency to deluxe a car by merely adding more chrome.

Generally speaking, I think instrument boards are inclined to be overdone. They are sometimes poorly organized and are apt to be confusing to the operator. I fully realize the sales aid provided by a very dramatic instrument board, but I believe a more dramatic and interesting effect could be achieved by removing some of the unnecessary ornamentation and concentrating the attention on a few of the really important instruments. *From a paper by Carl W. Sundberg, Sundberg-Ferar, presented at the annual technical convention of The American Society of Body Engineers Inc.*

Our Waning Fuel Reserves

FUEL reserves in the United States are being consumed at an alarming rate. If present trends continue, indications are that our reserves of natural gas and petroleum will become exhausted before the end of the century. Only coal appears to be in appreciable supply, with estimated reserves sufficient for approximately 2000 years at existing rates of consumption. On an equivalent heat basis more oil and gas are used per year than coal. Consequently, a depletion of our fluid fuel reserves would materially affect the economy of the nation.

The gas turbine appears to possess many attractive potentialities for relieving this serious situation. In conventional mining practices much coal

is left underground after operations cease, because beyond a certain point only a marginal return would be realized. Experiments under sponsorship of the Bureau of Mines are now in progress, however, involving burning of the coal remaining in such abandoned mines underground. The air required for this combustion can be supplied by a gas turbine, and the gases leaving the mine can, in turn, be burned in the turbine to supply useful power.

It is reasonable to expect that within the next decade power may be produced by nuclear fission. In this process, instead of burning a conventional fuel, heat would be obtained from a radioactive reactor. Heat thus produced could be imparted through a heat exchanger to a gas turbine cycle from which power is obtainable. The gas turbine has the unique advantage of being the most direct and simplest means of utilizing such energy. Uranium or thorium could be the "fuel" of the reactor, thus extending the reserves of conventional fuels. Production of power by this means would be another step, wherein a gas turbine is employed, toward fuel conservation, but it should not be assumed that the supply of uranium and thorium is inexhaustible, as such is not the case.

Use of Coal Should Be Stepped Up

Since our solid fuel reserves are in much greater abundance than those of oil or gas, wise utilization would be in the direction of developing efficient equipment capable of using the former in preference to fluid fuels. Most central power stations use coal, as do most steam locomotives. Now, under the sponsorship of the Locomotive Development Committee an efficient coal-burning gas turbine locomotive is being developed.

In power plants employing steam turbines and diesel engines much energy is thrown away in the exhaust because of the difficulty of recovering such low grade heat and putting it to a useful purpose. In a gas turbine, however, the exhaust gases possess a relatively high temperature so that energy is readily recoverable from them. Such hot gases can be used to make steam which in turn could be employed to heat buildings or for various processes.

In the event of exhaustion of our fluid fuel reserves the most obvious remedy, based on present knowledge, for domestic and other heating would be to use coal. It is possible, however, that synthetic fuel processes will be sufficiently advanced by that time so that the convenience of fluid

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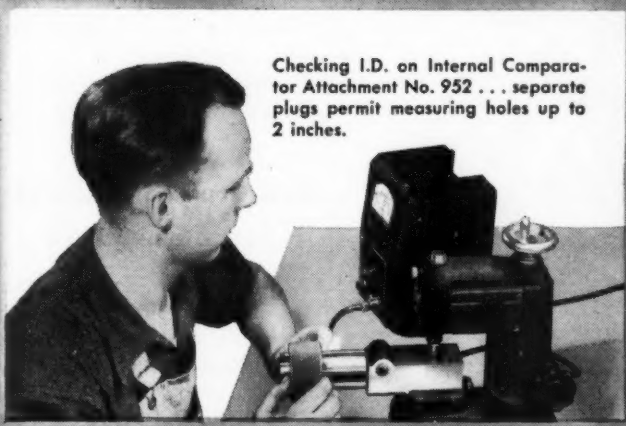
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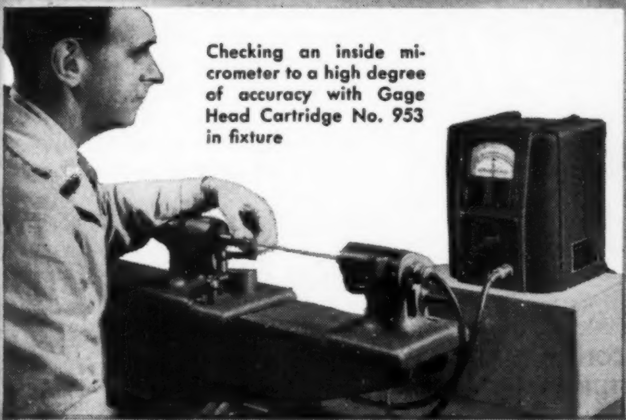
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Checking gage size with Gage Head Cartridge. Complete freedom from differences in "touch" of different operators.

With this equipment, you can gage a variety of small precision parts on a mass production scale so fast and so accurately that costs drop. No special skill needed. Savings over slower, more tedious methods quickly pay for this equipment.

When used as a part of a Quality Control Program, this Brown & Sharpe Electronic Equipment offers even greater savings. It permits closer, faster coordination of inspection with production . . . making possible a substantial reduction in rejects, wasted time and effort.

This modern gaging equipment is designed and constructed with characteristic Brown & Sharpe precision. For complete information on its unique features, write Brown & Sharpe Mfg. Co., Providence 1, R. I., U. S. A.

UNIQUE FEATURES

- Separate amplifier isolates heat-producing elements.
- True linear response permits accurate setting for entire scale with only 1 master block.
- Single amplifier may be used to serve several testing instruments.
- Normal line voltage fluctuations do not affect readings.
- Simple turn of graduation selector provides accurate readings in any desired increment from .0001" to .00001".
- Simple, self-checking device on external comparator . . . no gage blocks needed.

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fuels can be economically retained. Before the end of the century solar heating will be used to a much greater extent than at present, and it is even reasonable to expect that homes may be heated by electricity. Rapid progress being made in the power generation field would indicate such a possibility. *From a paper by Dr. John T. Rettaliata, dean of engrg., Illinois Institute of Technology, presented at the United Nations Scientific Conference on the Conservation and Utilization of Resources, at Lake Success, N. Y.*

How Transport Aircraft Can Be Improved

CAN we build an experimental office on the ground which vibrates as constantly and relentlessly as an airplane cabin? Can we place you and many other people in this vibrating cell for ten hours at a stretch—and have you eat your vibrating meals in it—and go to your vibrating bathroom in it—and keep your vibrating feet on the floor as well as your vibrating seat on the chair—for ten hours—and see how you like it? Can we see if it doesn't make you tired and depleted, like a ten-hour flight across our airways?

Can we determine that if you had a choice of a vibrating office or a vibrationless one that you would choose the vibrationless one? Can we build airplanes where the floors don't vibrate, the trays don't vibrate, the bathroom doesn't vibrate, the seats don't vibrate? Not knowing the answer to this research experiment, I merely hazard a guess that the public might find a vibrationless trip enjoyable.

Those of us who have been pilots hesitate to admit it, because it may not sound manly, but there is a definite emotional experience in seeing endless space on all sides—of experiencing the constantly changing views and vistas of the never-duplicated cloud formations. Psychiatrists, I believe, recognize the "flight from reality" as a definite human tendency—healthy, until it goes too far.

But today, these exciting views of the "wild blue yonder" are principally reserved for the crew in the front end. The most favored passenger (by the window) is given a head-sized round or square porthole, from which to catch an oblique glance of space, from a slightly-cocked neck angle. He must obtain his vicarious pleasures of seeing nature with the same general technique as peeking through a knothole in a nudist colony fence.

Can't you design windows that are

two or three or four times larger than the present ones? Can windows also extend downwards from the traditional height of a window ledge—down to the level of your ankles, perhaps, so you can see?

Is it absolutely necessary, after you have finished your great and tedious experiments in the most modern design of aircraft wings, to then place these wings directly in line of vision from the windows, and thereby deprive the window of its intended function—visibility? Or can the wing be out of the way?

The cross section of a passenger cabin may be roughly circular. An aisle runs down the middle. Two seats are on either side of the aisle. What happens when the cabin grows? Does the diameter of the cross section double? Are there four seats on each side of the aisle? Five seats on each side of the aisle? Even with larger windows this may be no good, if we research it far enough. Two seats by a window is some sort of human maximum for a deluxe or high-class service.

How About Seats in Tiers?

Can't we keep the cabins narrow, with two seats on each side of the aisle, but with three or four levels, if we are to have a transport which carries scores of people? Something mysterious happens when you put a hundred people all in the same cabin. Perhaps research will tell us what it is. Anxieties flare up more quickly in a mob than in a small group. The heaving shoulders and pale green discoloration of one upset passenger reflects in the unconscious reactions of ninety-nine other passengers when they are all together. Putting people on two, three or four different floors will keep the size of each group down, as well as insure better visibility.

Speed is the principal reason for air travel. Berths and other similar weight-consuming gadgets are, in the long run, only temporary expedients offered to distract the passenger because of our inability to get him there fast enough. When we have to drag along and take all night and part of the next day to get a passenger from New York to London, we may have to apologetically offer him an opportunity—at an extra fare—to take off his clothes and slip on his pajamas, to relieve the monotony. When we can take him from New York to London in five hours, or six, he won't have time to get undressed after we get him in the air, feed him, give him something to read, show him the disappearing American continent, and turn his head to see Ireland popping up in

front of him. Looking ahead to 1965, we suspect that anything that reciprocates, vibrates, and drags along at 400 miles an hour may be quite old fashioned. *From a paper by Harold R. Harris, vice president and general manager, American Overseas Airlines, presented at the SAE National Aeronautic Meeting in Los Angeles.*

Air Transport Supremacy Going To Britain

AMERICAN transport planes will soon lose the world-wide supremacy which they have possessed exclusively for so many years. With its new type gas-turbine and jet airliners Britain is winning its bid for universal air leadership. Many of our latest medium-range, piston-engine commercial aircraft are already obsolete. In the civil aircraft field America is repeating the same mistake made in World War II in military aircraft, when we permitted the British to conduct the development work on jet engines while our facilities and engineering talent were used for the mass production of conventional piston engines. As a result we lagged behind the British in the jet engine field, and spent several years in approaching their accomplishments. Even today we do not have jet engines with power ratings as high as the British.

The British are working on two types of commercial transport planes. One type, the turbojet, has gas-turbine engines where the high velocity exhaust gases from the engines propel the airplane by means of jet propulsion. This is the same method that is used in all of the latest high-speed military aircraft. The other type, the turboprop, has gas-turbine engines whose principal function is to drive a propeller, with some residual energy in the exhaust gases also furnishing a jet propulsion effect. Inherently a turbojet plane is faster than a turboprop type, because the efficiency of the propeller falls off at high speeds. Both turbojet and turboprop planes are capable of greater speeds than planes driven by piston engines with propellers. From a fuel consumption standpoint, however, the advantage is in the reverse order.

The de Havilland Comet is the most notable example of British efforts in the field of turbojet transports. The Comet is a four-engine, 36-passenger plane with a cruising speed of 500 miles per hour and a present range of 2000 miles at an altitude of 40,000 feet. After further modifications to

improve range it is intended to fly from London to New York in six hours. This is approximately half the time now required with conventional transports. Flight tests on the Comet have revealed it to exceed its cruising speed and altitude specifications. Fourteen Comets are on order for the British Overseas Airways Corporation, and two for the Ministry of Supply. Besides the Comet, the British have under development two other turbojet and four turboprop transports.

Why Tubojet Is Superior

There are many reasons why a turbojet transport is superior to the conventional piston engine type for medium ranges up to approximately 1000 miles. Admittedly the fuel consumption of the jet is higher than that of the conventional engine, but the much greater speed capabilities of the former result in lower total operating costs per mile. This derives from the fact that fuel cost is only part of the picture. There are other costs, common to all types of airplanes, such as crew salaries, depreciation, insurance, maintenance, etc., which have a greater influence on total cost. This latter type of cost in dollars per hour does not vary greatly between jet and piston-engine planes, with the result that the faster jet plane shows lower costs on a per-mile basis.

Additional advantages accrue when it is considered that for a given volume of traffic fewer jet planes are required due to their greater speed, consequently reducing investment. Furthermore, because of the purely rotary motion of the turbojets compared with the reciprocation of piston engines, the jet airliner is virtually vibrationless, thus materially enhancing passenger comfort. Its cabin noise level is also greatly reduced. Maintenance time on the turbojets should also be lower. The Air Force states that only five man-hours are required for a complete change of turbojet engines compared with 150 man-hours for piston engines.

There is not a single turbojet or turboprop transport now under construction in this country. Even if such construction were started today, it would still be at least three years before commercial operation could be realized. Undoubtedly progressive airlines in this country will of necessity be purchasing foreign jet transports in the interest of continuing the policy of rendering to the public the best type of service available.

From a technical and production

standpoint the United States is better qualified to develop turbojet transports than any other country. It is unreasonable to expect, however, that private industry, even though convinced of the desirability, could undertake such a program without federal assistance. The British developments have been under government subsidy. It would appear logical to employ some military appropriations for such a purpose, as transports are certainly used to a large degree in the normal conduct of wartime operations. If the United States is ever to regain its eminent position in the transport field positive action should be taken, and soon. *From a talk by Dr. John T. Rettaliata, dean of engineering, Illinois Institute of Technology, presented at a recent meeting of the American Institute of Electrical Engineers in Cincinnati.*

Standards in Design

THE design engineer is usually the man who initially specifies the commodities which procurement must ultimately obtain. His interest lies in the development of the products assigned him. His concentration, naturally, is in the accomplishing of his objective, in each case, if possible, an accomplishment that will be hailed as a truly significant milestone in the art.

In proceeding with this development, he will usually make good use of the well-established, known standards that are available to him. Normally, however, he has a flare for the individualistic touch, and specials frequently appear in the picture, interesting to be sure, but questionable when viewed in the light of practical need. It, therefore, becomes a problem to guide the engineer in the full effective use of standards and at the same time avoid discouraging the free and unrestrained use of his imagination, which is required if we are to progress. The washing machine on the drawing board today is the procurement activity and the production line of tomorrow.

In talking with our purchasing people, I constantly stress the importance of realizing this fact and, therefore, urge that all possible effort be made to influence the use of standards in the early development stage. Cooperative effort between Engineering and Purchasing can greatly simplify the work of procurement when the production plans are made. *From a paper by V. de P. Goubeau, director of materials, RCA Victor Div., Radio Corp. of America, presented at Thirty-First Annual Meeting of ASA.*

Beginning of An Important Product Improvement

No machine can be better than its bearings! Bearings are basic . . . to long life, to dependability . . . to the performance and reputation of the product. In the product planning stage be sure to investigate the advantages that SEALMASTERS exclusive combination of features can give to your product.

It's no accident that a great many of the foremost machine manufacturers are specifying SEALMASTER Ball Bearing Units. Pillow Block (shown above), Cartridge, Flange, Flange-Cartridge, Take-Up and Hanger Units, as well as many special types, are illustrated in the SEALMASTER catalog. Have your secretary write for it today.

Bearing Division
STEPHENS-ADAMSON
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5 Reasons Why You're Better Off with SEALMASTERS

1. **Permanently Sealed**
Felt-lined steel flinger rotating in labyrinth, prevents entry of dirt and retains proper amount of lubricant.
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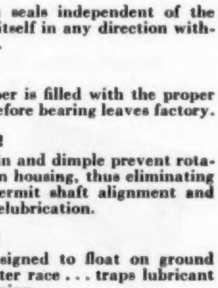
Bearing unit, with seals independent of the housing, can align itself in any direction without seal distortion.
3. **Prelubricated**
The bearing chamber is filled with the proper amount of grease before bearing leaves factory.
4. **No Housing Wear!**
Patented locking pin and dimple prevent rotation of outer race in housing, thus eliminating housing wear . . . permit shaft alignment and position unit for relubrication.
5. **Floating Retainer!**
Ball retainer is designed to float on ground inner surface of outer race . . . traps lubricant and prevents churning.

Invite the SEALMASTER engineer to give your company the full story on SEALMASTER bearings. If you have a special bearing problem . . . put it up to him for recommendation.



FACTORY REPRESENTATIVES AND DEALERS IN ALL PRINCIPAL CITIES

SALES AND SERVICE PERSONNEL

THE POSITION of district sales manager of the magnesium aluminum division of Utica Radiator Corp. has been filled recently by **Maurice G. Steele.** For the past ten years Mr. Steele has been connected with the Kent Co. of Rome, N. Y., first as factory manager and later as vice president in charge of engineering for an affiliate of the Kent Co., the Kent Electric Corp. Previous to that he was associated with Revere Copper & Brass Inc., Rome, N. Y., as manager of their copper radiation division and later as technical advisor of the Baltimore division.

♦

Worthington Pump and Machinery Corp. has recently announced the appointment of **A. M. Shaw** as assistant manager, pump and compressor merchandising division. Mr. Shaw will be responsible for the development of pump and compressor sales to original equipment manufacturers, and his activities will come under the supervision of **J. O. Glenn,** manager of the division.

♦

Reorganization of the sales and engineering department of the Fawick Airflex Co. Inc., manufacturers of industrial clutches, was recently announced. **John V. Eakin,** formerly southwestern district manager, was appointed assistant sales manager, and **J. S. Walsh** was named assistant chief engineer. **Frank C. Warth** was appointed southwestern district manager with offices in Fort Worth, Tex., and **C. H. Lay** was made Chicago district manager with offices in Evanston, Ill.

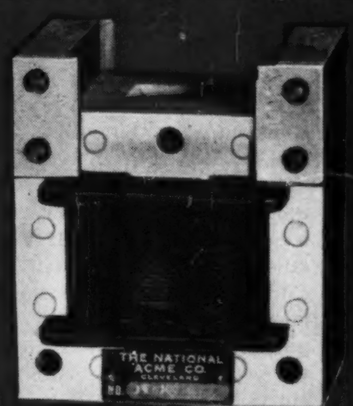
♦

Announcement has been made of the appointment of **John L. Sinclair** as district manager of the mechanical goods division of the Goodyear Tire & Rubber Co. He has been associated with this division since 1924.

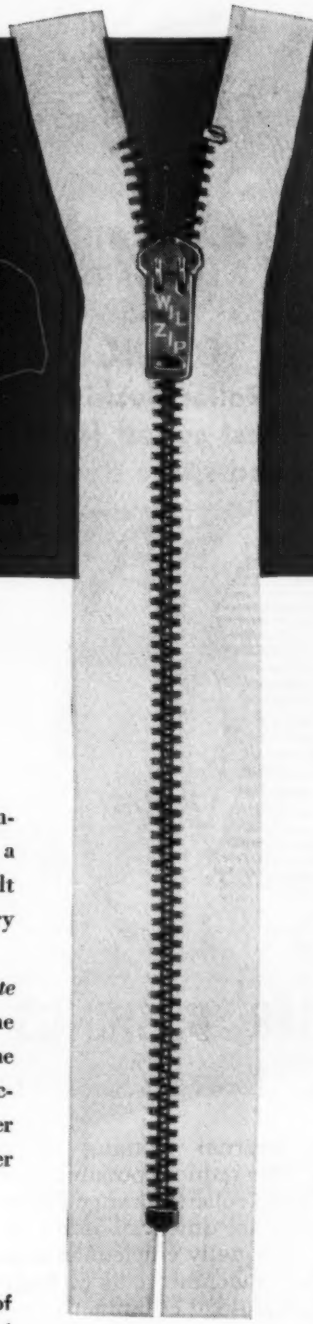
♦

The appointment of **F. P. Taugher** as manager of engineering for the industrial control division of Westinghouse Electric Corp. in Buffalo was announced recently. Mr. Taugher, who has been engineering and service manager in the company's New England district for the past five years, has wide experience in the applica-

QUICK-ACTING! ACCURATE!



-Type Namco Solenoid, 5/8" stroke, 21½ lbs. pull, as used on slide fastener chain machine.



SLIDE-FASTENER MANUFACTURER SAYS NAMCO IS ONLY SOLENOID DEPENDABLE FOR HEAVY-DUTY, PRECISION SERVICE ON CHAIN MACHINE

NAMCO
"STELLITE"-WELDED
SOLENOIDS

Namco Solenoids are precision built and custom-engineered to the individual job—to provide a push (or a pull) of up to 25 pounds. They are built in a variety of sizes and mountings to meet every commercial application.

When you want positive, automatic, *accurate* operation—even under conditions of extreme heavy-duty service—let Namco Solenoids do the job. They'll activate clutches, motor drives, electric switches, hydraulic valves and other power controls, even replacing fractional horsepower motors, in many cases.

Complete standard range of Namco "Stellite"-Welded Solenoids is illustrated and tabulated in Engineering Bulletin EM-46A. Ask for your copy.

Close ones don't count—slide fasteners don't work unless they're *on the nose.* That's why dead-accurate operation of the chain machine that makes them is so vital. This machine, running *continuously* at 2500 strokes per minute, is the key unit controlling production in the entire plant.

Here's what Plant Engineer R. A. Proud, of the Wilson Fastener Division of Talon, Inc., has to say: "We have come to rely on Namco Solenoids. Experience has taught us they are *the only ones* that will stand up under the punishing action of constant vibration, energizing instantaneously at the *exact* interval required to feed in the wire stock."

While such whole-hearted endorsement is gratifying, it is by no means unique. In industry after industry, manufacturers everywhere are coming to rely on Namco Solenoids for heavy-duty, precision service. Is there a tough job in your plant we can help you lick?

ELECTRICAL MANUFACTURING DIVISION

The NATIONAL ACME CO.

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Acme-Gridley 4-6 and 8 Spindle Bar and Chucking Automatics • Single Spindle Automatics • Hydraulic Thread Rolling Machines • Automatic Threading Dies and Taps • The Chronolog • Limit, Motor Starter and Control Station Switches • Solenoids Centrifuges • Contract Manufacturing

tion, installation and servicing of control equipment in the field, in addition to a broad background in control design work. **M. L. Gardner** has succeeded Mr. Taugher as engineering and service manager of the New England district, with headquarters in Boston. He has been a Westinghouse employee since 1924.

♦

Fred L. Hopf has been named general manager of Dynamatic Corp., Kenosha, Wis., manufacturers of variable-speed drives, punch-press drives, brakes for oil well draw works, dynamometers, etc. Dynamatic Corp. is a subsidiary of Eaton Mfg. Co., Cleveland. Having been with Dynamatic Corp. since 1933, he has been closely connected with the engineering department as sales engineer and also as sales manager of the corporation since 1948.

♦

The Standard Transformer Co. of Warren, O., has announced the appointment of **F. L. West,** 106 Temple St., Avon, N. Y., as its representative for western New York state. He offers a complete line of transformers for distribution, power, metering, and testing.

♦

Harry C. Platt has been appointed vice president of engineered castings division of American Brake Shoe Co. Formerly works manager, he has been with the company since 1942. **William H. Starbuck** has been made vice president of the company's Kellogg Division. He was previously assistant general sales manager, serving in the sales departments of other divisions before joining the Kellogg division.

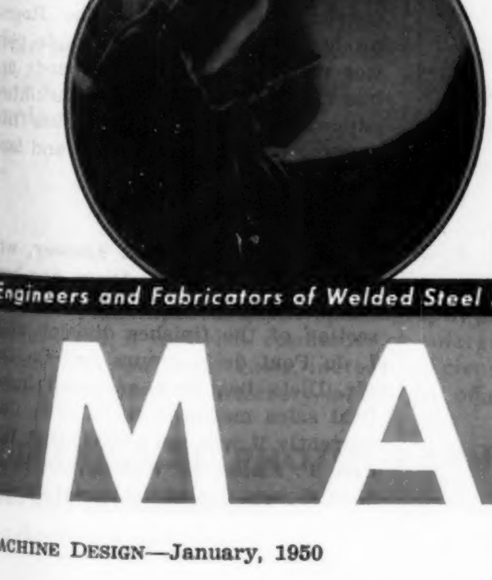
♦

Formerly covering the New York metropolitan area, **Leonard A. Ashworth** will now represent the Continental Screw Co. in Maine, New Hampshire, Vermont and Connecticut. **Robert A. McCormick** has left the company's office staff to take over the New York metropolitan area, and **Paul R. Flemming** is a new Continental representative in the state of New York and a portion of Pennsylvania.

♦

According to an announcement by Chicago Vitreous Enamel Product Co., **Wesley L. Dinsmore** and **William L. Donaldson** have been appointed district managers. Mr. Dinsmore, who has been a service engineer in the eastern territory, has been appointed district manager of the New England territory. He joined the Chicago Vit

Steel-Weld FABRICATION



This marine steam engine crankcase and frame is another excellent example of fine workmanship in Steel-Weld Fabrication . . . a job requiring a high degree of accuracy in fabrication, and precision in machining operations . . . it is typical of heavy machinery parts and assemblies produced by Mahon for many customers throughout the country. You, too, will find in the Mahon organization an unusual source for welded steel in any form . . . a source with complete production and machining facilities, backed by a staff of competent design engineers and craftsmen from whom you may expect a smoother, finer appearing job, embodying every advantage of Steel-Weld Fabrication.

THE R. C. MAHON COMPANY
DETROIT 11, MICHIGAN

Engineers and Fabricators of Welded Steel Machine Bases and Frames, and Many Other Welded Steel Products

MAHON

organization in 1947, in the capacity of service engineer. Mr. Donaldson, who was formerly a service engineer in the Ohio territory, has been made district manager of the east central states.

♦

Stephen C. May has been named vice president and general sales manager of Blackmer Pump Co., Grand Rapids, Mich. He formerly served as consulting sales counselor with the Auto-Soler Co., in sales executive capacities with Pacific Mfg. Corp. and Iron Fireman Mfg. Co., and as vice president and sales manager of Darling Valve & Mfg. Co.

♦

The Carpenter Steel Co. has announced the appointment of **Omar V. Greene** as New England sales manager, with headquarters in Hartford, Conn. He succeeds **Wynn F. Rossiter**, who has been made assistant to the vice president to devote his time to special work. **John W. Thompson**, formerly manager of alloy steel sales, has been named manager of sales development to succeed Mr. Greene in this capacity at the company's main office in Reading, Pa.

♦

William J. McQuillan has been promoted to the position of branch manager, Pittsburgh office, Morse Chain Co., division of Borg-Warner Corp. Prior to joining the sales staff of Morse Chain Co. in 1948, Mr. McQuillan was associated with the sales department of H. H. Robertson Co., after having spent several years in the engineering department of Blaw-Knox Co.

♦

Rogers Corp., Manchester, Conn., has announced the appointment of **John H. Beach** as New York state representative. Mr. Beach's territory will comprise all of New York state, excluding the metropolitan New York district, and Erie, Pa. The Rogers products he will handle include plastics molding compounds, boards and blanks; Duroid; electrical insulating papers and boards, and Rogers' fabricating services for fibrous and laminated phenolic materials.

♦

Succeeding **Henry E. Lackey**, who is retiring, **Joseph B. Dietz** has been made manager of the industrial sales section of the finishes division of E. I. du Pont de Nemours and Co. Inc. Mr. Dietz has been assistant industrial sales manager since 1945. Concurrently it was announced that **William P. Fisher Jr.** was appointed assistant manager of industrial sales in

Wilmington, Del.; **William E. Kreuer** was transferred to Chicago as assistant manager of industrial sales; **Richard N. Sanger** was appointed industrial sales manager in Cleveland; and **Walter S. Woods II** was named industrial sales supervisor at Cleveland.

♦

A new corporation manufacturing permanent-mold castings and jointly owned by the Barnes Mfg. Co. of Mansfield, O. and the Non-Ferrous Die Casting Co. Ltd. of London, Non-Ferrous Perma Mold Inc. has appointed **S. E. Gregory** as general sales manager for the corporation. An authority on permanent-mold casting of all non-ferrous alloys, Mr. Gregory was associated for twelve years with the Non-Ferrous Die Casting Co. Ltd. While with the British company he served in various capacities in tool and engineering design, foundry production, and sales engineering.

♦

The Westcott Chuck Co., Oneida, N. Y., has announced the appointment of **Herman H. Hobelmann,** 744 Harrison St., San Francisco, Calif., as its district representative in the states of California, Oregon, and Washington and in the Reno district of Nevada.

♦

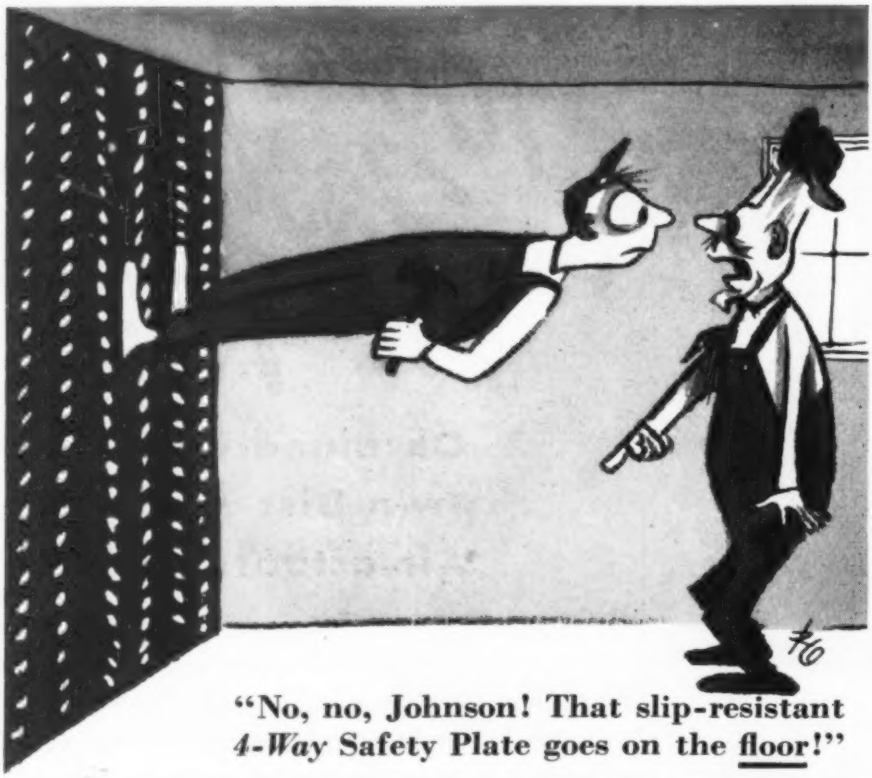
Homer A. Goddard Jr. has been made assistant general manager, industrial marketing, for Gulf Oil Corp. In his new post, created as part of the firm's reorganized domestic marketing program, Mr. Goddard will be responsible for direct sales of Gulf products to industrial plants and organizations. He joined the company in 1932 as industrial lubrication engineer with the Pittsburgh division, later becoming superintendent of industrial sales for the division, and then assistant division manager in charge of industrial sales.

♦

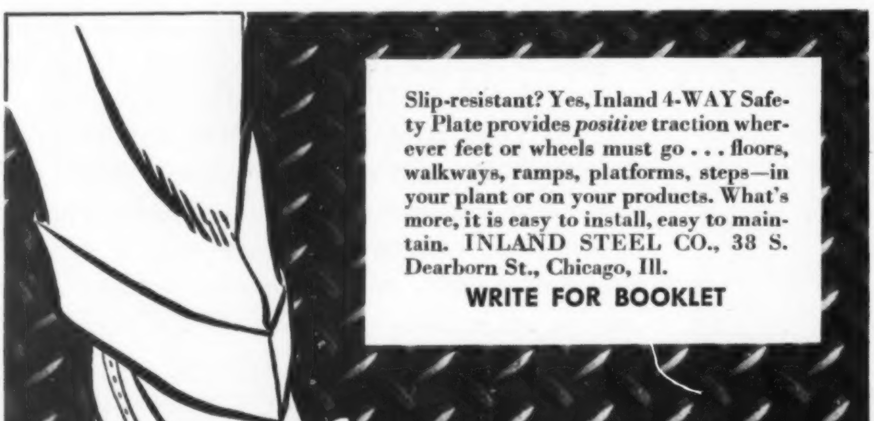
Diamond Chain Co. Inc., Indianapolis, has announced the appointment of **Howard W. King** as West Coast district manager with headquarters in San Francisco. Mr. King has a background of twenty-one years with Diamond Chain, the past fourteen years as district sales representative in Chicago.

♦

J. H. Goodspeed, formerly in the Chicago sales office of Titan Metal Mfg. Co., Bellefonte, Pa., has been placed in charge of the company's St. Louis sales office at 817-819 Arcade Bldg., St. Louis 1, Mo. His new territory comprises Missouri, Iowa, Kansas, and southern Illinois.



"No, no, Johnson! That slip-resistant *4-Way* Safety Plate goes on the floor!"



Slip-resistant? Yes, Inland 4-WAY Safety Plate provides *positive* traction wherever feet or wheels must go . . . floors, walkways, ramps, platforms, steps—in your plant or on your products. What's more, it is easy to install, easy to maintain. INLAND STEEL CO., 38 S. Dearborn St., Chicago, Ill.

WRITE FOR BOOKLET

Inland 4-WAY* SAFETY PLATE

STOCKED BY LEADING STEEL WAREHOUSES

*Reg. U. S. Pat. Off.

SALES NOTES

SERVING the territory of Detroit and the southern half of Michigan, as well as Fulton, Williams, Ottawa and Lucas counties in northern Ohio, the **Milton A. Meier Co.,** 816 New Center Bldg., Detroit 2, Mich., has been appointed distributor for pig and ingot products of the **Reynolds Metals Co.**

♦

Ward Leonard Electric Co., Mount Vernon, N. Y., has announced the establishment of their St. Louis branch office at 4030 Chouteau Ave., St. Louis 10, Mo. Charles B. Durling Jr. is district manager. At the same time it was announced that Storer and Schemm, Cincinnati representative of the Ward Leonard Electric Co., has changed its name to **Sheldon Storer and Associates,** retaining its location in the Transportation Bldg., 307 East Fourth St., Cincinnati 2, O.

♦

Manufacturer of Le-Hi high and low pressure hose couplings, **Hose Accessories Co.** of Philadelphia has announced that its West Coast factory warehouse is now located at 1320 Santa Fe Ave., Los Angeles 21, Calif. The new warehouse carries a complete stock of hose couplings, clamps, nipples and related accessories. Availability of this large stock in Los Angeles will provide improved service for Le-Hi customers in the entire Pacific Coast area.

♦

The **Standard-Thomson Corp.,** aviation and automotive parts manufacturer, has leased the Andrews Building at 128 Tulsa Lane, Dayton, O., for use as a warehouse and shipping center. The building will provide the company with 30,000 square feet of additional floor space and office facilities.

♦

Announcement has been made of a change in the sales and manufacturing organization of **Jas. P. Marsh Corp.** Sales of the newly acquired Electrimatic line of refrigeration control valves and regulators are to be handled by the **Electrimatic Co.;** pressure gages, dial thermometers and

GRAMIX self-aligning bearings and U.S.G. carbon brushes

help make new SPEEDRILL a "fistful of po

The six GRAMIX self-aligning bearings sealed into the rugged little Speed-Way power-drill have a lot to do with its amazing long life and performance. And the pair of U.S.G. carbon brushes in this speedy tool assures absolute balance of brush pressure on the commutator at 1000 R.P.M.

GRAMIX bearings and specialty parts are die-pressed from powdered metal, *finished to size without machining.* They are sufficiently porous to retain lubricant and thus eliminate the need of regular lubrication. U.S.G. carbon brushes are available for any type motor from 1/100th to 100 H.P. Write for your copy of the new GRAMIX and the new U.S.G. Brush catalogs.

THE UNITED STATES GRAPHITE COMPANY • SAGINAW, MICHIGAN
DIVISION OF THE WICKES CORPORATION

related products will be handled by **Marsh Instrument Co.**; and Marsh traps, valves and other heating specialties will continue to be handled by **Marsh Heating Equipment Co.** Each of these organizations will be designated as sales affiliates of Jas. P. Marsh Corp.

♦

Direction of sales activities for welding electrodes and equipment of the **A. O. Smith Corp.,** Milwaukee, has been transferred from the north central district, Chicago, to the welding and electrode and equipment division, Milwaukee. This move is designed to bring sales and technical service to customers closer to the research and production facilities. All sales activities for these products will be directed from the Milwaukee office under the supervision of J. T. Pritchard, division manager. L. F. Vonier is general sales manager for the division.

♦

St. John X-Ray Laboratory, Califon, N. J., has received a 350 millicuries source of Cobalt 60, a radioactive material to be used for industrial radiography in place of radium. Since Cobalt 60 has a smaller focal spot size than the equivalent amount of radium, better results can be obtained by the use of this new radioisotope. A separate building has been designated for the exclusive use of Cobalt 60, and it can also be used on any field job.

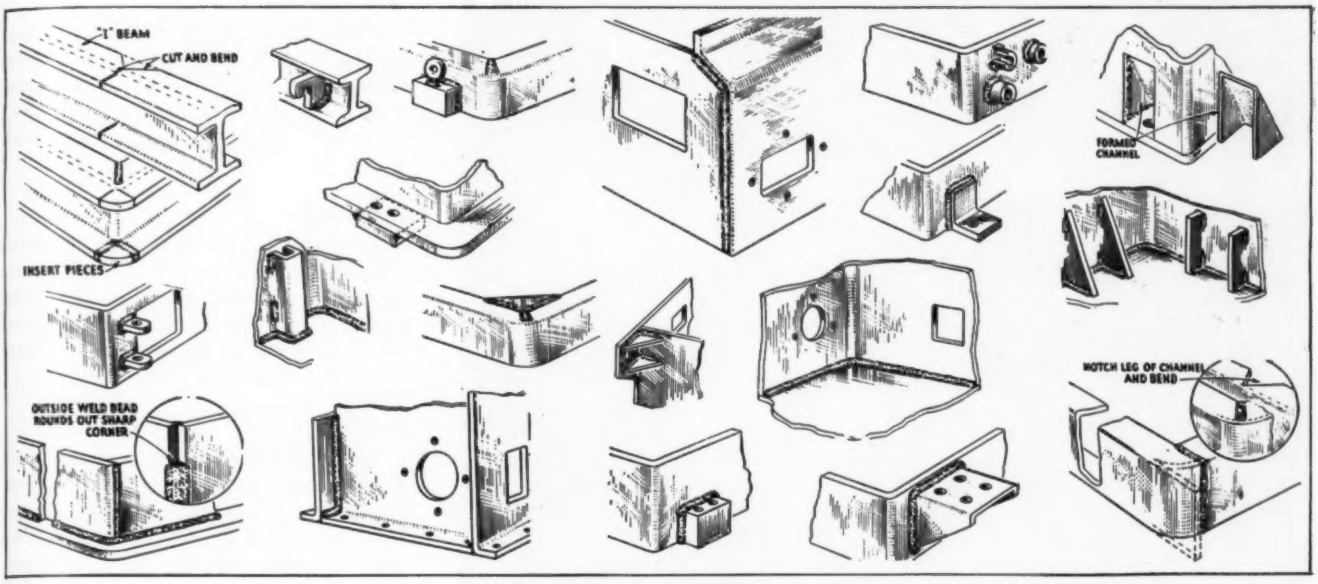
♦

Canadian representatives for **Cambridge Wire Cloth Co.,** the **R. D. Travers Co.** has moved into new and larger quarters at 117 Charlton Ave., West Hamilton, Ontario. The additional facilities were necessary to keep service in step with demand for Cambridge woven wire conveyor belts, industrial wire cloth and special wire fabrications.

♦

American Manganese Steel Division of American Brake Shoe Co. has announced the appointment of the **Whitehead Metal Products Co. Inc.** as distributors of its complete line of Amsco welding products, consisting of various types of rods and electrodes for hard facing and repair work and including the newly developed tungsten carbide rods. The Whitehead Co. operates warehouses in major cities in five Eastern states including New York, New Jersey, Pennsylvania, Massachusetts and Maryland.

Ideas on Designing Bases for Greater Rigidity at Less Cost



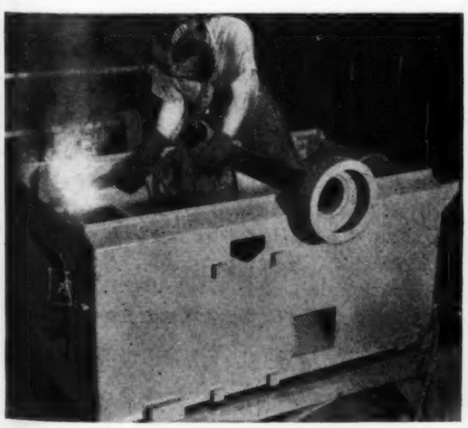
Examples of simple design details for fabricating more durable machinery bases at less cost with arc welding.

Machinery bases built from welded steel possess more than twice the rigidity per pound than cast iron. By fabricating with arc welding, manufacturers are able to incorporate many unique design features at less cost since most component parts are readily cut and formed from standard steel shapes and plate and then clamped in simple fixtures for fast, easy arc welding.

Suggested above are but a few of many design ideas that can be used to simplify the construction of machinery bases, and at the same time, achieve greater rigidity, clean, modern appearance at a remarkably low cost.

In a great many cases, plain structural shapes like steel bars, channels, "I" beams and simple plate are used almost entirely. Where metal forming equipment is available, component parts can be bent to shape, thereby minimizing both fit-up and welding. Many components can also be pre-drilled and tapped on small, high speed equipment, eliminating the need for heavy, slower operating machine tools.

Design improvements or changes to suit customers' needs are more easily accomplished with welded design. Costs and delays of pattern changes are eliminated, thus cutting down overall production time and speeding manufacturing schedules.



Rigid All-Welded Base *for planer. Side members are sheared plate reinforced with stiffeners. Courtesy Porter Machinery Company, Grand Rapids, Mich.*

Clean, Modern Styled Appearance *is made possible with structural shapes and plate. Base has totally enclosed reservoir for coolant storage.*

● More detailed analysis for low-cost design of machinery bases is contained in the "Procedure Handbook of Arc Welding Design and Practice." Price is only $1.50 postpaid in the U.S.A.; $2.00 elsewhere.

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MEETINGS AND EXPOSITIONS

Jan. 16-19—

Plant Maintenance Show to be held in the Cleveland Auditorium, Cleveland, Ohio. Additional information may be obtained from Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 23-26—

Institute of Aeronautical Sciences. 18th annual meeting to be held at the Hotel Astor, New York, N. Y. Additional information may be obtained from Society headquarters, 2 East 64th St., New York 21, N. Y.

Jan. 23-27—

American Society of Heating and Ventilating Engineers. Fifty-sixth annual meeting and the Southwest Air Conditioning Exposition to be held in Dallas, Texas. A. V. Hutchinson, 51 Madison Ave., New York 10, N. Y., is secretary of the society and Charles F. Roth, Grand Central Palace, New York 17, N. Y., is exposition manager.

Feb. 8—

American Society of Mechanical Engineers. Machine Design division panel discussion meeting on "The Influence of Customer Demands and Suggestions on the Design and Performance of Capital Goods Machinery" to be held at Hotel Schroeder, Milwaukee. Additional information may be obtained from T. F. Eserkaln, 2250 N. 62nd St., Wauwatosa, Wis.

Feb. 12-16—

American Institute of Mining & Metallurgical Engineers. Annual meeting of the Iron and Steel Division, the Institute of Metals Division and the Extractive Metallurgy Division in connection with general meeting of AIME to be held at the Hotel Statler, New York, N. Y. Ernest Kirkendall, 29 West 39th St., New York 18, N. Y., is secretary.

Feb. 27-Mar. 3—

American Society for Testing Materials. Spring meeting to be held at the Hotel William Penn, Pittsburgh, Pa. Robert J. Painter, 1916 Race St., Philadelphia 3, Pa., is assistant to the secretary.

Mar. 14-16—

Society of Automotive Engineers. Passenger car, body and production

meeting to be held at the Book-Cadillac Hotel, Detroit, Mich. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Mar. 28-31—

Society of the Plastics Industry. Fourth national plastics exposition to be held at the Navy Pier, Chicago, Ill. William T. Cruse, 295 Madison Ave., New York 17, N. Y., is executive vice president.

Apr. 4-8—

National Production Exposition sponsored by the Chicago Technical Societies Council will be held at the Stevens Hotel, Chicago, Ill. John C. Toohy, 176 W. Adams St., Chicago 3, Ill., is exhibit manager.

Apr. 10-14—

American Society of Tool Engineers. Industrial exposition to be held at Convention Hall, Philadelphia, Pa. Additional information and advance registration forms may be obtained from Harry E. Conrad, executive secretary, 10700 Puritas Ave., Detroit 26, Mich.

Apr. 17-19—

Society of Automotive Engineers. Aeronautic meeting and aircraft engineering display to be held at the Statler Hotel, New York, N. J. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

May 8-12—

American Textile Machinery Exhibition to be held in Atlantic City Auditorium, Atlantic City, N. J., under the sponsorship of the National Association of Textile Machinery Manufacturers. E. Kent Swift, Whitinsville, Mass., is president.

June 11-16—

American Electroplaters' Society. 37th annual convention to be held at the Hotel Statler, Boston, Mass. The Fourth International Electrodeposition Conference in collaboration with the Electrodepositors' Technical Society of England will be held at the same time. Additional information may be obtained from American Electroplaters' Society, 473 York Rd., Jenkintown, Pa.

April 24-27—

American Management Association. 19th national packaging exposition to be held at the Navy Pier, Chicago, Ill. Edward K. Moss 330 West 42nd St., New York 18, N. Y., is public relations director.

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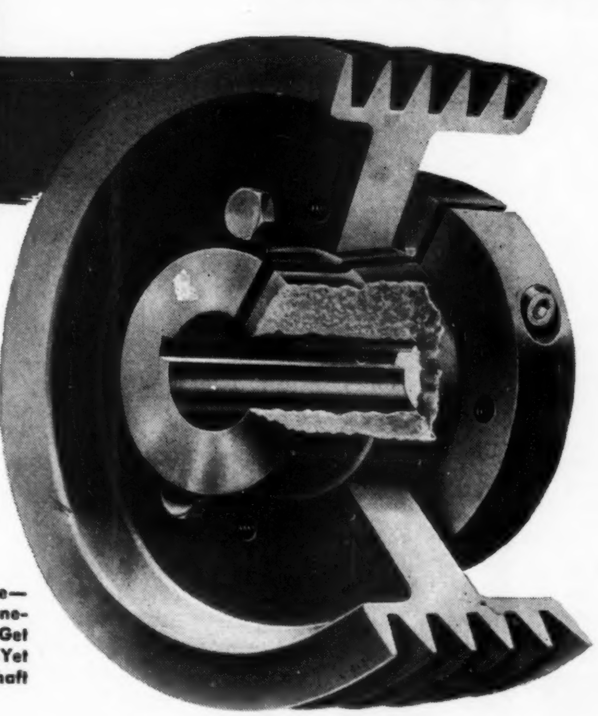

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GAS FURNACE. Input capacities: 84,000, 112,000 and 140,000 Btu per hour. Comes in 2 assembled packages; heating section is 23 in. high, 50 in. long, blower section fits against heater or can be installed remote from heater. AGA approved for zero clearance to combustibles on top, bottom and back of heat exchanger. The Lennox Furnace Co., Marshalltown, Iowa.

INDUSTRIAL GAS BURNER. Designed for natural or mixed gas. Gas burner mounting plate also has lugs for mounting oil burner when both oil and gas are to be used. Gas unit operates with natural or forced draft. Iron Fireman, Portland, Oreg.

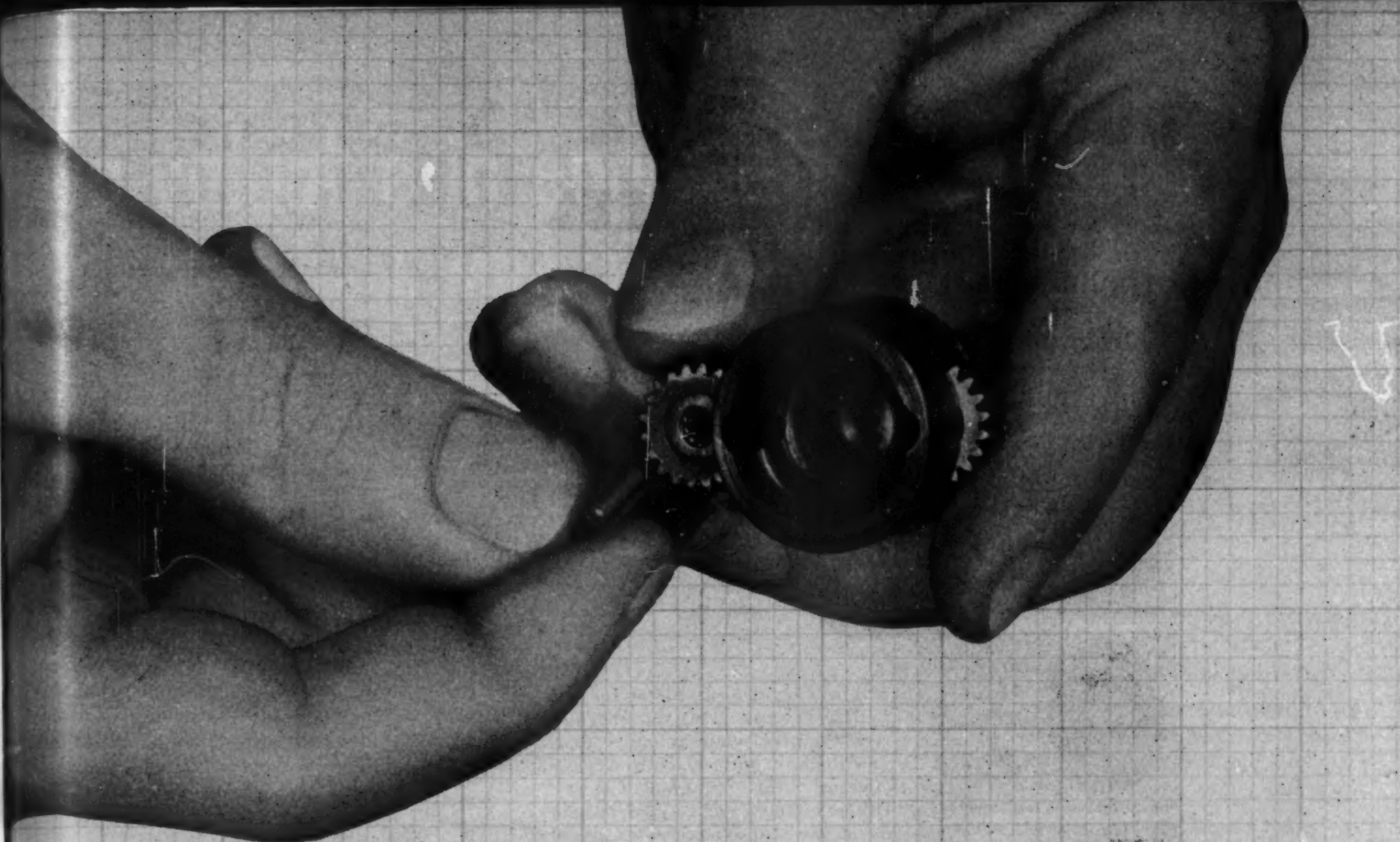
FORCED-AIR GAS FURNACES. Capacities: 160,000 and 200,000 Btu per hour input. Narrow "clamshell" heat extractors force combustion gases into thin layer, giving max efficiency in heat exchanger. Single port type of burner used, 1 for each 40,000 Btu of input. Burner adjustable for manufactured, mixed and natural gas. Lennox Furnace Co., Marshalltown, Iowa.

Heat Treating

PREHEATER. Portable unit mixes, dries and preheats plastic powders. Feeds hot thermoplastic granules directly into hopper of injection and extrusion machines. Capacity, 10 lb per hour. Aluminum tumbler driven at 3 rpm by 100-watt gearmotor. Tilting hinged tumbler mounting plate forward empties tumbler. Heating accomplished with 375-watt lamp mounted over tumbler. Drying speeded with 1/20-hp fan mounted on lamp bracket. Miskella Infra-Red Co., Cleveland, O.

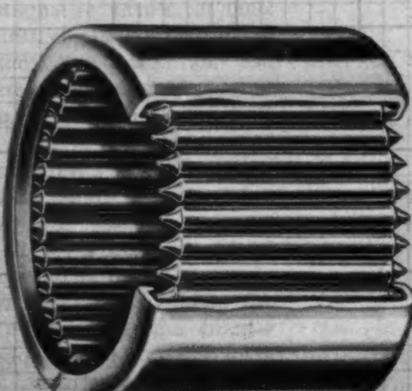
INDUCTION HARDENER. For continuous, r-f induction hardening of cylindrical parts at feed rates to 6 in. per second. Automatic loader feeds pieces to a horizontal rotating scanner through a feeder unit. Work hardened by passing through an induction coil and then cooled with spray quench. Uniformity of case depth is obtained by controlling feeds. Any desired hardness pattern can be obtained by adjustment of electronic timing circuits. Westinghouse Electric Corp., Pittsburgh, Pa.

HEAT TREATING FURNACE. Bench type



TORRINGTON NEEDLE BEARINGS

contribute to compact design



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unit for tool and die hardening and tempering and as furnace for batch production runs of small parts. Max temperature, 1850 F. Dimensions, 10 in. wide, 8 in. high, 18 in. long. One model permits manual setting of power input at any point from 5 to 100% of full rating. Temperatures held at close limits with control pyrometers. Other model designed for pyrometer operation but without auxiliary equipment, for work requiring less critical temperatures. Cooley Electric Mfg. Corp., Indianapolis, Ind.

Manufacturing

POWER HACK SAW. Cuts material 2⅝ by 2⅝ in. and permits angle cutting of 1⅝ in. stock. Equipped with ¼-hp, 1725-rpm motor operating machine at 150 strokes per minute. Max cutting stroke, 3½ in. Standard equipment includes 8-in. blade and extension bar accommodating 10-in. blade. Pressure relief on return stroke eliminates drag and unnecessary blade wear. Sales Service Machine Tool Co., St. Paul, Minn.

PORTABLE POLISHING TOOL. For rubbing and polishing metals, woodwork, linoleum, etc. Weight, 4¼ lb; free speed, 1300 rpm. Swivel side handle can be positioned at any position on tool for convenient operation. Balanced design prevents tool from tipping over. Includes die-cast aluminum frame, universal motor, alloy steel planetary gearing, and 6-in. diameter backing pad. Cummins Business Machines Inc., Portable Tools Div., Chicago Ill.

REVERBERATORY TILTING FURNACE. For melting of nonferrous metals, iron, ferro manganese, copper ferrite and malleable iron. Capacities, 500 to 2000 lb. Furnaces capable of producing 13 to 21 continuous melts per day. Each unit complete with burner, blower, motor and air ram. Bellevue Industrial Furnace Co., Detroit, Mich.

DOUBLE-CRANK PUNCH PRESS. Large bolster area Model 3036 rated at 30 tons capacity. Bolster area, 16 by 36 in.; ram area, 10 by 36 in.; press speed, 80 strokes per minute. All steel welded construction with 4-point engaging clutch. Standard stroke, 2 in.; max stroke to order, 4 in.; ram adjustment, 2 in. Press is single-geared using herringbone gears. Requires 3-hp motor operating at 1800 rpm. Diamond Machine Tool Co., Los Angeles, Calif.

HAND GRINDER. For light bench and finishing work, touch-up work on dies and molds, chain saw sharpening, etc. Weight, 12½ oz. Uses 1/40-hp universal motor operating on 115 volts. Pushbutton lock secures tools in place. Housing con-

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sists of 2-piece casting with all major parts mounted in one half for ease in cleaning or repair. The Dumore Co., Racine, Wis.

SPECIAL-PURPOSE CRANKSHAFT MACHINE. Drills, counterbores and taps 168 holes in aluminum crankcase. Consists of welded steel columns on each side of bed, each column carrying 2 standard Snyder units. Each unit has two 14-spindle heads and 2 additional tapping units on each column. Power is supplied by three 7½-hp motors at 1200 rpm, two 7½-hp motors at 1800 rpm, and two 5-hp motors at 1800 rpm. Total cycle time is 2 minutes, 19 seconds without time for loading and unloading, which is manual. Snyder Tool & Engineering Co., Detroit, Mich.

LATHE. Precision model with 1-in. collet capacity, 8½-in. swing. Features stepless spindle speeds from 32 to 2000 rpm. Entirely mechanical drive designed for changing spindle speeds with machine running. Electric tachometer continuously registers spindle speeds. Powered by 1-hp motor, machine has high torque at low spindle speeds because of 9 to 1 back-gear ratio. High speeds obtained using matched V-belts. The Wade Tool Co., Waltham, Mass.

PORTABLE GRINDERS AND POLISHERS. For light polishing, routing, brushing, grinding, buffing, drilling, sanding, etc. Machines deliver ¼-hp at 1750 rpm through 5-ft flexible shaft. Additional model available with speeds of 1000, 1750 and 3000 rpm, obtainable by plugging flexible shaft into different positions on gearbox. Both models mounted on casters. Stow Manufacturing Co., Binghamton, N. Y.

UNIVERSAL DIECASTING MACHINE. Small, low-cost, high-pressure hydraulic model. Two-stage, 10-hp pump delivers 43 gpm at low pressure stage and 10½ gpm at 500-psi stage. In either cold chamber model for aluminum, or gooseneck style for zinc, tin or lead. Automatic or manual controls. Dimensions: die plates, 22 by 18 in.; diameter of 4 tie bars, 2 in.; space between tie bars, 12 by 16 in.; and die opening, 8 in. Estimated 50 tons locking pressure through toggle mechanism. Produces aluminum castings to 2½ lb, zinc castings to 4 lb. Free cycle time of machine, over 1000 shots per hour. The Cleveland Automatic Machine Co., Cincinnati, O.

ADJUSTABLE DRILL HEADS. Multiple-spindle models for drilling holes from 0.013 to 0.1875-in. drill sizes. Two-spindle unit has close center

In a Diesel engine, in the boom of an excavator, or in the landing gear of an airplane, sleeve bearings and bushings are the equivalent of jewels in a watch. They must meet high precision standards.

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To produce the fine performance your specifications call for there can be no compromise at any point in manufacture. Our seven sleeve bearing manufacturing plants—the result of over 50 years' specialization—with extensive research and engineering facilities, can combine quantity, low cost and quality in bearings that will do your job well. Your inquiry is invited.

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HEAVY LOAD for big Diesels, power plants, etc.—bearings up to 27½" O.D., steel and bronze back.

BRONZE PARTS in many shapes, sizes; thrust washers, bushings; for many types of applications.

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FEDERAL-MOGUL CORPORATION FEDERAL Mogul *Silent* SLEEVE BEARINGS **11045 SHOEMAKER, DETROIT 13, MICH.**

spacing of 0.75-in., wide center setting of 2.062 in. Four-spindle model has close center setting of 1.218 in., wide center setting of 2.375 in. in a bolt circle; in square pattern, close centers are 0.875-in., wide centers are 1.6875 in. Spindles on both heads will operate on any setting in full 360-degree range. Aiman's Machine Tool and Engineering Co. Inc., Pendleton, Ind.

PNEUMATIC DIE CUSHION. Operates on shop air line. Adaptable to large straight-side presses. Ring holding pressures vary from 15 to 25 tons. Heavy-duty pin pressure pads made in sizes from 28 by 36 in. to 34 by 48 in., custom made to fit press bed opening. Dayton Rogers Manufacturing Co., Minneapolis, Minn.

AUTOMATIC DRILLING UNIT. Hydraulically controlled. Amount of rapid approach, rate of feed and overall length of stroke varied by external adjustment. Handles 1/32 to ¼-in. tools with standard collet chuck. With special chuck machine handles ⅜-in. tools in soft materials. Govro-Nelson Co., Detroit, Mich.

SHEARS. High-speed power shears for light-gage sheet metal. Operates continuously at 125 strokes per minute on mild steel to 20 gage. Cutting edge fully visible for accurate cutting to layout lines. Fabricated from electrically welded steel plate with bronze bearings and ways. Niagara Machine & Tool Works, Buffalo, N. Y.

LATHE. Heavy-duty, 16-in. model. Heavy spindle with 24 speeds mounted on 3 bearings. Hardened and ground gears give speed range of 15 to 1000 rpm in standard models. Multiple-disk, wet type clutch and brake combination running in oil used to control spindle. Shifter mechanism allows shifting directly from one speed to any other speed. Tailstock has handwheel offset 50 degrees toward operator, has 2-speed action for slow drilling feeds and rapid advance. The Springfield Machine Tool Co., Springfield, O.

GRINDER. Twin-wheel, wet or dry tool grinder for carbide, stellite and high-speed tools. Both roughing and finishing wheels on same machine. Independently tilting tables for each wheel adjustable between 15 degrees above and 30 degrees below horizontal. Grinding spindles and coolant pump are V-belt driven and fully enclosed. Cranks give independent wheel adjustment for each wheel. The Standard Electrical Tool Co., Cincinnati, O.

CIRCULAR DIVIDING MACHINE. For production ruling of precision circular

scales on instruments and machine tools. Lines can be ruled on any horizontal, beveled or cylindrical surface at speeds to 60 lines per minute. Spacings accurate to plus or minus 15 seconds of arc. Max size of work handled, 20 in.; max diameter on vertical rulings, 10⅝ in.; length of lines, 0.02 to 0.5-in. The Gaertner Scientific Corp., Chicago, Ill.

WHEEL CRUSHER UNIT. Portable. For crush-forming grinding wheels. Roll arbor mounted with heavy-duty roller bearings, unit equipped with ⅛-hp, 115-volt, 60-cycle, single-phase motor. Standard unit used to form grinding wheels to 1¼ in. wide. Wider wheels formed by addition of longer roll spindle. Jerpbak-Bayless Co. Inc., Chagrin Falls, O.

PRESS. Capacity, 5 tons. Weight without motors, 300 lb. Includes roller bearings in flywheel, cam actuated brake, and nonrepeating safety mechanism. Frame is of semisteel unicast construction. Sales Service Machine Tool Co., St. Paul, Minn.

PORTABLE ELECTRIC DISHWASHER. Does not require permanent plumbing or electrical connections, resulting in reduced installation costs. Rubber inlet hose connects to hot water faucet, drain hose discharges into sink. When not in use, machine rolls out of the way. Dishwasher has same capacity as deluxe automatic machine but is manually controlled. Dimensions: height, 36 in.; diameter, 23½ in. General Electric Co., Bridgeport, Conn.

MOWER. Horizontal-blade type machine powered by 2-hp, vertical-shaft, air-cooled motor. Use of vertical engine shaft eliminates clutch and belt drive. Carriage mounted on 3 pneumatic tires. Height adjustable from 1½ to 3 in. Weight of manually propelled machine, 87 lb. Moto-Mower Co., Detroit, Mich.

AIR COMPRESSOR. Portable. Capacity, 500 cfm at 100 psi. Unit starts as low-compression gas engine, is shifted to full diesel operation after short warm-up period. Speed regulator adjusts compressor to lowest practical working speed to hold desired pressure. Ingersoll-Rand Co., New York, N. Y.

CONTOUR GRINDER. Truforming machine reduces grinding time on 8-in. circular saws from 10 to 2½ minutes. Grinding wheel covers entire saw area to be shallow tapered, both grinding wheel and saw rotating during grinding. Outer edge of blade ground to 0.086-in., tapered to 0.058-in. one inch from edge,

then to 0.069-in. at point 1⅝ in. from center of saw. Truing and working cycles of machine are automatic, with pushbutton control. Thompson Grinder Co., Springfield, O.

Plant Equipment

AUTOMATIC WEIGHER. Electronically-controlled unit for weighing to 5 lb. Has bucket capacity of 270 cu in., operates with bags, cartons, and glass and metal containers. Feeder handles tea leaves, spices, coffee, cereals and other materials too light to pass through gravity feeders. Electronically-controlled, high-speed valve closes when exact weight is reached. Speeds to 25-30 per minute on 1-lb charges with accuracy of 1/16-oz possible. Shock absorber removes vibration caused by opening and closing of bucket door. Can be set for automatic or manual operation. The Automatic Scale Co. Inc., New York, N. Y.

PORTABLE DUST COLLECTORS. Heavy-duty units with capacities of 450, 900 and 1800 cfm at high air velocity. Can be set up to service individual machines in isolated locations, eliminating costly pipe runs. Unit consists of motor, exhauster, centrifugal precleaner and steel wool filter after-cleaner. Collectors are fire proof; exhaust stack connections furnished at top or any side of units. The Kirk & Blum Mfg. Co., Cincinnati, O.

COAL DRYER. For spreader-stoker units. Mixture of air and furnace gas enters dryer at about 1000 F, dry and preheats coal. Coal is sufficiently preheated to materially reduce time for ignition, reducing amount of fine coal carried into boiler passages. Iron Fireman, Portland, Oreg.

ELECTRIC HAMMER. Model No. 10 has ⅝-in. diameter drilling capacity.Use of light-weight selenium rectifier makes unit self-contained. Since rectifier is now incorporated in handle casting, operator plugs 110-volt a-c power line directly into hammer. Syntron Co., Homer City, Pa.

Power Equipment

STANDARDIZED CONTROL CENTERS. Containing motor-starter units for centralized control of groups of machines. Feature plug-in type disconnects on line side for added safety. Units are 20 in. wide, 20 in. deep, with heights in multiples of 14 in. Fit standard indoor structures either 76 or 90⅜ in. high Can also be mounted back-to-back in same structures. Panels can be locked in tilt-out position, disconnected from line. Connectors automatically connect line terminals

when starters are inserted. Units baffled from each other for added short-circuit protection. Westinghouse Electric Corp., Pittsburgh, Pa.

LIGHTWEIGHT ENERGIZER. Supplies power suitable for medium requirements of low-voltage d-c power for aircraft, airports and aircraft testing stations. Two-unit, 2-bearing, self-ventilated motor-generator set rated at 265 amp, 7.5 kw. Carries 500-amp overload for 1 minute. Regulated voltage ranges are 13-17 volts and 28-35 volts. Regulation within plus or minus 2½% over entire range. Induction motor is squirrel cage type, 220, 440 or 550-volt, 2 or 3-phase, 60-cycle. General Electric Co., Schenectady, N. Y.

Materials Handling

PORTABLE BELT CONVEYOR. For use as floor-to-floor conveyor and stacker. In belt widths of 6 to 36 in., lengths of 12, 15, 18, 20 and 24 ft. Either slider or roller bed construction, furnished with power-driven tail feeder and wheel, roller gravity section or nosed-over delivery. Screw-jacks in raising mechanism permit locking at any desired height. Conveyor rides on 2 stationary and 2 swivel 5-in. casters. Floor locks maintain desired location. Sage Equipment Co., Buffalo, N. Y.

STACKER. Hand-operated model with load capacity of 1000 lb. Raises 24 by 26-in. platform from 6 to 58 in. Platform locks in any position, cannot settle under load. Overall dimensions: height, 70 in.; width, 32 in.; length, 42 in. Platform movement accomplished through hand crank, roller chain, bevel gears and screw. Provision made for rapid platform movement under light loads. Weight, 328 lb. The Hamilton Tool Co., Hamilton, O.

AUTOMATIC PACKAGE FEEDER. Positions bundles for required number of ties in standard wire-tying machines. When used for newspapers, feeder handles 24 single-tied bundles, 20-in. high, per minute. Control switch regulates mechanism for required number of ties; electrically-controlled roller drive and air-actuated pusher position the bundle for tying. Feeder stops if wire breaks, preventing jamming. Curtis Development & Manufacturing Co., Milwaukee, Wis.

Testing and Inspection

DYNAMOMETER. Fractional-horsepower unit for determining speed-torque curves, fuel or energy consumption, efficiency curves, max running torque, etc., of both driven and driving machines. Has continuous absorption rating of ¾-hp at 1725

rpm and intermittent rating of 1½ hp. Max safe speed, 4000 rpm. Includes 9¾-in. Chatillon dynamometer scale graduated in 0.1 lb subdivisions, control panel, etc., and optional indicating tachometer equipment. General Electric Co., Schenectady, N. Y.

X-RAY THICKNESS GAGE. Designed to measure thickness of cold-rolled steel strip ranging in thickness from 0.0050 to 0.1196-in., in steps of 0.0001-in. Gage compares amount of X-ray absorption from 2 beams identical in intensity, 1 through sample of exact thickness to be rolled and another through strip being rolled. Results are interpreted visually. Electrical and radiation safeguards provided for personnel. Westinghouse Electric Corp., Pittsburgh, Pa.

TIRE BALANCER. For truing tires for roundness and then balancing. Equipped with two ¼-hp motors, 1 for rotating tire and other for operating rotary knife. Pantograph templates control arc of knife across tread. Handles all size tires. O. K. Rubber Welding Mfg. Co., Littleton, Colo.

UNIVERSAL TESTING MACHINE. Capacity, 60,000 lb in 2 ranges; max capacity range graduated in 10-lb units and a 12,000-lb range graduated in 20-lb units. Hydraulic loading unit separate from indicating and control unit. Tensile or compression load applied by integrated piston and elevating cage mechanism. Loading controls give infinitely variable speeds between 0 and 2 in. per minute. Vertical distance between gripping heads, 1 to 18 in.; clear lateral space between columns, 10 in. Baldwin Locomotive Works, Philadelphia, Pa.

VIBRATION TEST STAND. For 2-dimensional vibration of loads to 500 lb. Specimens tested under constant amplitude or constant force conditions. Semiportable machines require no concrete base; all vibratory forces are absorbed in suspended unit. Turntable to obtain third direction of vibration available. Frequency range, 3 to 60 cps; amplitude range, 0 to ½-in. Amplitude controlled by gas pressure displacing liquid in vibration generators. L. A. B. Corp., Summit, N. J.

Transportation

TRAILERS. For carrying earth-moving equipment. Folding gooseneck lowers to ground to form loading ramp. When trailer is loaded, power-operated winch on truck raises gooseneck to towing position. Capacities, to 100 tons. Martin Machine Co., Kewanee, Ill.